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# MACHINERY:

VOL. 3. No 12. PUBLICATION OFFICE  
411-418 PEARL STREET  
NEW YORK CITY. AUGUST: 1897.

A PRACTICAL JOURNAL FOR MACHINISTS AND ENGINEERS  
AND FOR ALL WHO ARE INTERESTED IN MACHINERY:

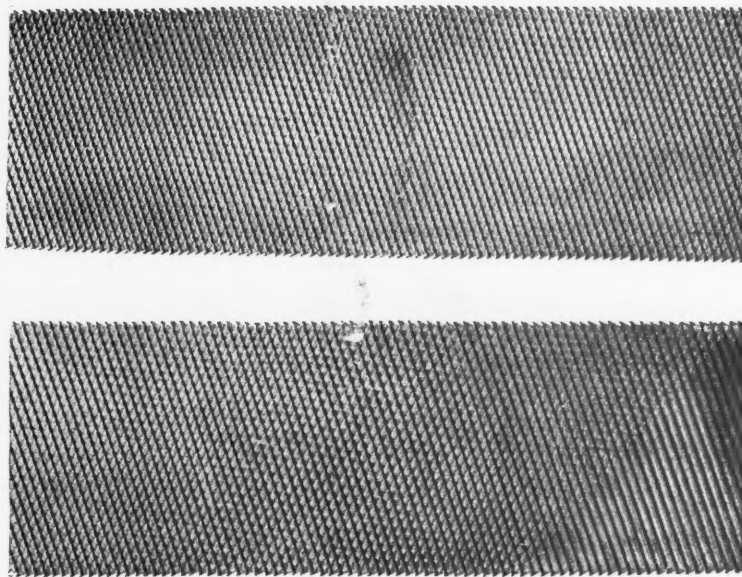
## Some Common Sense about Files.

Nearly every mechanic knows that the difference between a hand cut and a machine cut file consists in the irregular spacing of the teeth--the difference of a thousandth of an inch, more or less.

For some reason which has never been discovered, files with this irregular spacing--which up to a recent date was a peculiarity of those made by hand--took hold of the work better than those with regular spacing, so that quite naturally file hands have come to prefer the hand cut to the machine cut, and a good many manufacturers found it economical to furnish the former, although they cost more. Within a year or two, however, file manufacturers have been experimenting with machine made files having irregular spacing, and Mr. Alfred Weed, of the Arcade File Works, has produced what is known as the Weed Increment, cut by a patent flexible chisel which

throws up an exact reproduction of the hand cut file; so exact indeed that in a well known eastern shop the Arcade Increment Cut Files, with the name obliterated, were recently handed

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out to the men who had previously been using hand cut files, and not one of them knew the difference. They went ahead with the Arcade Files and obtained as good results in every way as they have previously done with the more expensive hand cut files. And, in fact, there is no difference either in appearance or in results--the only difference is in the cost. Thousands of dozens of the Arcade Files are now being sold in Great Bri-

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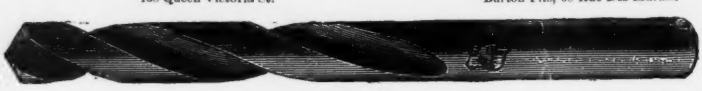
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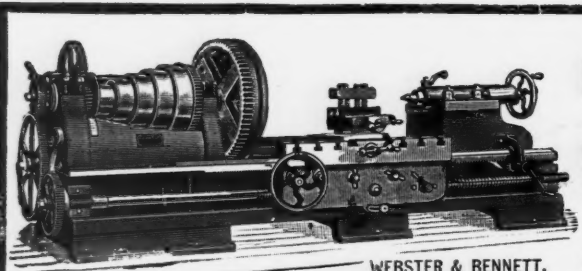
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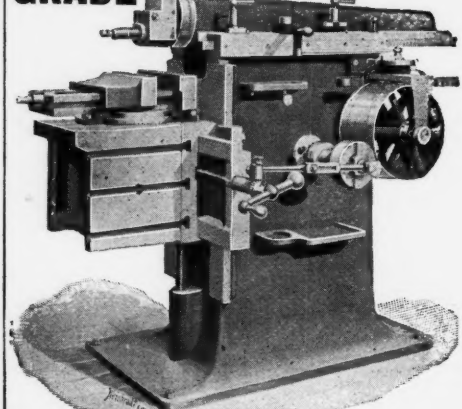
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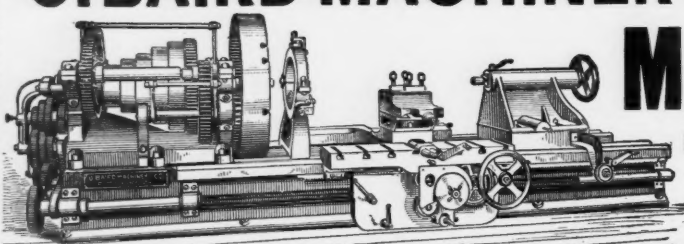
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# MACHINERY.

VOL. 3.

August, 1897.

No. 12.

## NOTES FROM THE BUILDERS' IRON FOUNDRY SHOPS.

It is unusual to find a firm located a mile from either railroad or boat, and competing successfully in the line of heavy castings and machine work, often for distant points of the country. Such, in fact, is the situation and condition of the Builders' Iron Foundry, of Providence, R. I. This plant is one of the two that are turning out the heaviest castings made in New England, and the shops, which, together with the foundry, go under the firm name of the Builders' Iron Foundry, are always engaged in work of a heavy character.

The equipment of the shops is modern and but few old machines are to be seen. At the present time the work is being pushed on a Government contract for 35 12-inch, steel breech-loading mortars, a line in which this concern has had a great deal of experience. Many interesting examples of machine work occur in the construction of these mortars, and I trust our readers may hear more of them and of the tools and machines used at a later day. By reason of the variety of work, both large and small, which is done at the Builders' Iron Foundry, and of the many original ideas that have been developed there, it is a place of unusual interest.

The regular drawings, as far as possible, are detailed on small sheets, 9 x 12 inches (standard size), blue-printed, mounted on tin, and filed away in the tool-room, to be taken out on check, only. Plus and minus limits are placed on the drawings, showing the degree of accuracy expected of the

### HISTORICAL NOTES.

In marked contrast to the equipment are the buildings themselves. They are simply a series of additions made from time to time, to an original foundry building, as business demanded it. The original building was erected as early as 1822, and probably earlier, before the days of railroads and steamboats. This fact explains the unsatisfactory location of the works.

In 1832 the High Street Foundry was established on the same site, and in 1853 this gave way to the Builders' Iron Foundry, which was incorporated at that time. From this time up to 1862 the chief work of both the foundry and shops was iron work for

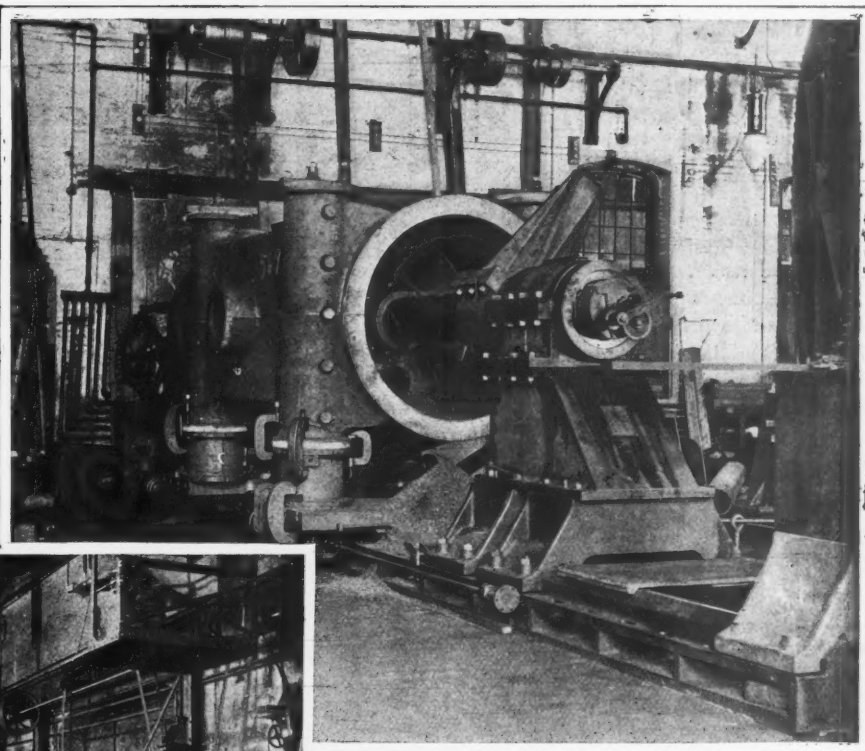


FIG. 1.—BORING AN ENGINE CYLINDER..

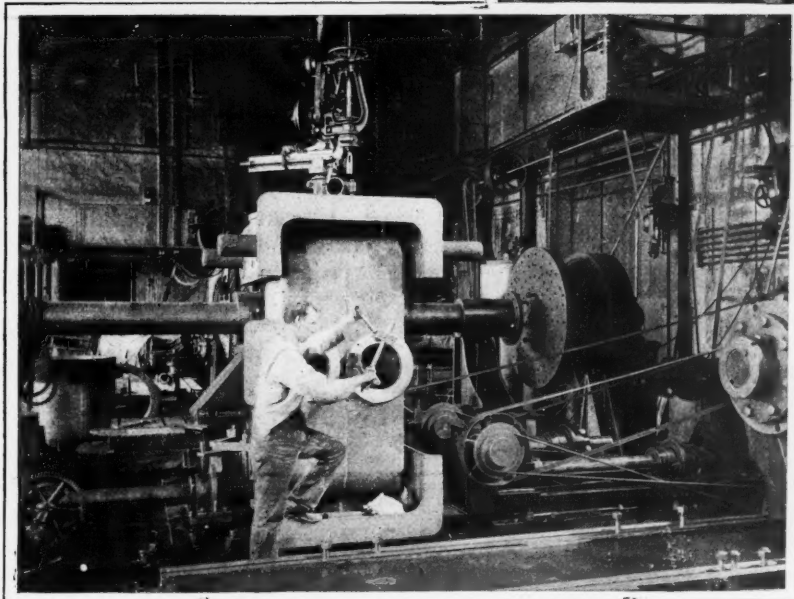


FIG. 2.—A PORT-BORING MACHINE.

workman. These limits take care of the "running fit" and "forcing fit" sizes. To further insure good and accurate work, the system of inspection by operations by a separate inspection department is in vogue. The tool-room is supplied with an elaborate collection of limit gauges and plugs.

In view of the intimate connection that the Builders' Iron Foundry has maintained with the mechanical life of Providence during the past 75 years, or more, a few historical notes will be of interest.

builders' use, and stoves. In this year the manufacture of heavy guns was begun for use in the civil war.

It is said that the first planer ever run in New England was set up in these shops. It would take work about three feet square and of considerable length. It had wooden shears, hand-chipped Vs and most of the iron work was imported. It was screw driven, and later the Builders' Iron Foundry engaged in the manufacture of planers, using the same drive, one of which is still running in the shop. It is shown in Fig. 4. Mr. William Hicks, a veteran Providence mechanic who runs the planer has been employed continually by this firm for 33 years.

Prominent among the mechanics who were connected with these works at one time or another in its early days were George H. Corliss, Edward Bancroft, of the original firm of Bancroft & Sellers, of Philadelphia, and N. T. Greene, the inventor of the Greene engine, now made by the Providence Steam Engine Co. Corliss came here about 1840 to have a sewing machine built. It was of his own invention, for use in sewing leather, and he worked upon it himself as a mechanic. Later he went in the employ of the Hope Iron Works, Providence, and was engaged

in building slide-valve engines having wooden frames and connecting-rods, and fly-wheels with rims left unturned. The latter position was the beginning of his steam-engine career. Bancroft left Providence in 1848, and with William Sellers, established the firm of Bancroft & Sellers. He died in 1855, when the firm name became William Sellers & Co.

#### AN OLD LATHE.

At the time when the manufacture of guns for the civil war was contemplated, four lathes were built for the purpose, one of which is shown in Fig. 1, at work on a recent job of boring. The boring-bar and its outer support are new, but the bed and head-stock are the original ones. The latter is shown clearly in the background in Fig. 2. The spindle is driven by a belt and cone at right angles to it, the connection being through a pair of bevel gears and a pinion running in a large internal gear, which is seen back of the face plate.

This lathe has a long record of useful work to its credit, from facing a lot of bridge pier cylinders, 11 feet in diameter, to other work within its rated capacity of 48-inch swing.

#### A HOME-MADE TOOL.

The "double-spindle, port-boring machine" shown in Fig. 2 appears to be an exception to the general rule that in nine cases out of ten it does not pay to build tools at home, unless in the business. It is used for boring the valve seats in the cylinders of the engines, which are built here for the Rice & Sargent Company, of Providence, and which are an important factor in the work of the shops. The engines are of the four-valve, Corliss type, and two seats are bored at one setting. The records show that the finished diameters do not vary more than from .001 to .002 of an inch from end to end, or more than would be expected from the wear of the tool, owing to the large amount of scale that it necessarily comes in contact with in rubbing over the edges of the ports.

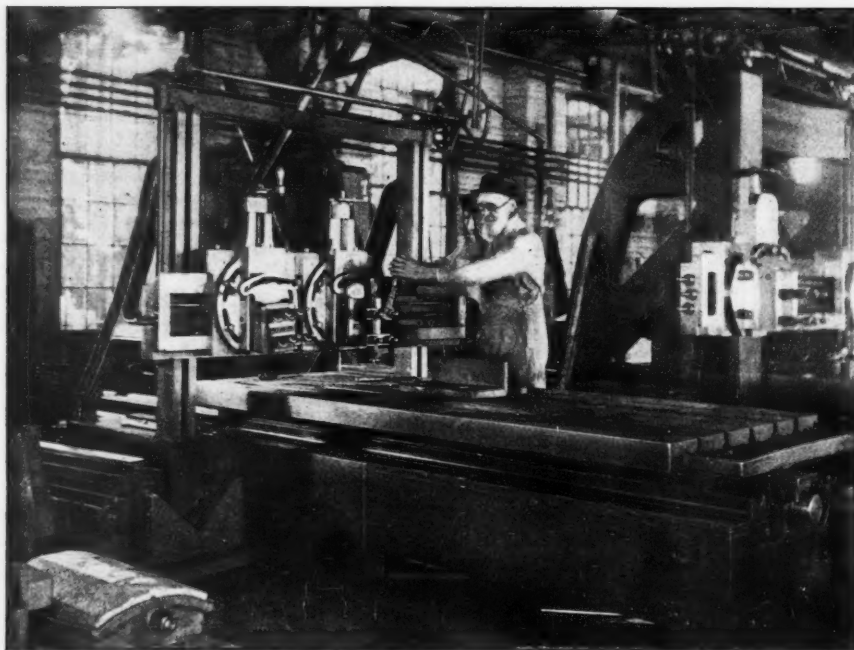


FIG. 4.—A VETERAN MECHANIC AND HIS PLANER.

As to the machine itself, its base consists of a floor-plate,  $7\frac{1}{2} \times 15$  feet. Upon this and arranged to be moved a greater or less distance apart, as required, are two feeding beds carrying the feeding and driving mechanisms and one end of the boring-bars. One bed is seen in the lower right-hand corner of the illustration, and more or less is visible of the two driving heads, the two bars and the steady rests for the outer ends of the bars. Each bar is

driven independently by belt from one counter-shaft, through the medium of a steel worm and bronze worm-wheel, the ratio of drive being 1 to 15. The feeding mechanisms, situated at the back ends of the feeding beds, are driven by ropes from the main heads, as plainly shown in the engraving. The bars are fed along bodily. Ample provision is made for a change of feed of



FIG. 3.—A PROVIDENCE LANDMARK.

from .02 to .7 of an inch. The height of the bars above the bed is a fixed distance, and shims are used for the different cylinders to bring them to the required height.

When an engine cylinder is being made ready for boring, a plate drilling jig is placed on its upper end and clamped through to the bed. A Dallett drill is then clamped to the center of the jig and the holes for the cylinder-head studs are drilled and tapped simultaneously with the boring, and often by the same man. This combination of operations, arranged in this particular way, makes an original and an effective method for machining engine cylinders, and is due to Mr. A. A. Fuller, the shop superintendent.

#### A MODERN SHAPER.

There are several modern machine-tools here that are noteworthy, but they are mostly at work on the mortars, and reference to them will be deferred. I will mention one, however, a shaper made by W. H. Warren, the radial drill man, which escaped notice in the shaper article in the issue of May, 1896. It is not a well-known machine, but it was built by a man who believes in good work and generous wearing surfaces to the extent of putting his views into practice, in this instance, at least.

It will be seen from the engraving that the connecting link between the crank and ram pulls instead of pushes during the cutting stroke and that a friction down-feed is provided. The transverse feed, however, is a positive instead of a friction feed, and causes the tool to drag during the back stroke. Apparently, also, the machine is less convenient to operate than the ordinary pillar shaper, and has a weaker drive than some shapers, though probably it is powerful enough. In spite of these facts, however, the shaper has every appearance of one that will out-wear most machines of its class by several years, and runs the quietest of any that I have seen.

L. G. F.

## THE POSITION OF DRAUGHTSMAN.

G. EDWARD SMITH.

There is a great diversity of opinion as to the relative position of a draughtsman and his rank in the engineering profession. As a usual thing his importance is greatly underestimated, not only by mechanics but by engineers themselves. As a class they receive but small praise; it is so easy, as I have said in a previous article, for the superiors to keep the credit and shift the blame. In the case of the shop-hands there is a certain jealousy of his position which causes them to try to depreciate the draughtsman's usefulness.

A mechanical draughtsman's value does not depend on his ability to produce the most artistic and finished pieces of pen and ink work, but upon his capabilities as a designer and as an adapter of ideas, requiring the combining of the best schemes of engineering practice with his own to suit the particular case in hand. In almost all cases his designs must be such that the cost of manufacture is as low as possible, and yet the machine or other article must be most efficient in its operation and easily manipulated.

As his work is principally brain work, a draughtsman should be ranked as a professional man, although I fear this opinion is not held by many of our egotistical friends of the other professions, who seem to feel that they only should be considered the brain-workers of the country. As a matter of fact the engineering pro-

seems to be an attempt made to keep each shop's or yard's practice for the use of its officials only. This was shown quite clearly by the address about fifteen months ago of the president of the Junior Engineers, in London, at their annual meeting. In this he was most strong in his criticism of the draughtsman, and said that a draughtsman had no right to take any notes of the practice of the company with which he is employed, as that practice belongs exclusively to the firm by whom it is worked up. This view may be taken, but it certainly is not a logical one, for if each firm were to keep to its own practice alone, how far advanced would we be in the arts and sciences? The first hundred years of iron moulding showed practically no advance in the art, and it was not until its secret became known that its development started in strides towards its present status. The result of individual firm practice would be that as its members aged, so their ideas would become covered with cobwebs and their work be getting far behind the times.

This brings us to the subject of keeping records of our prac-

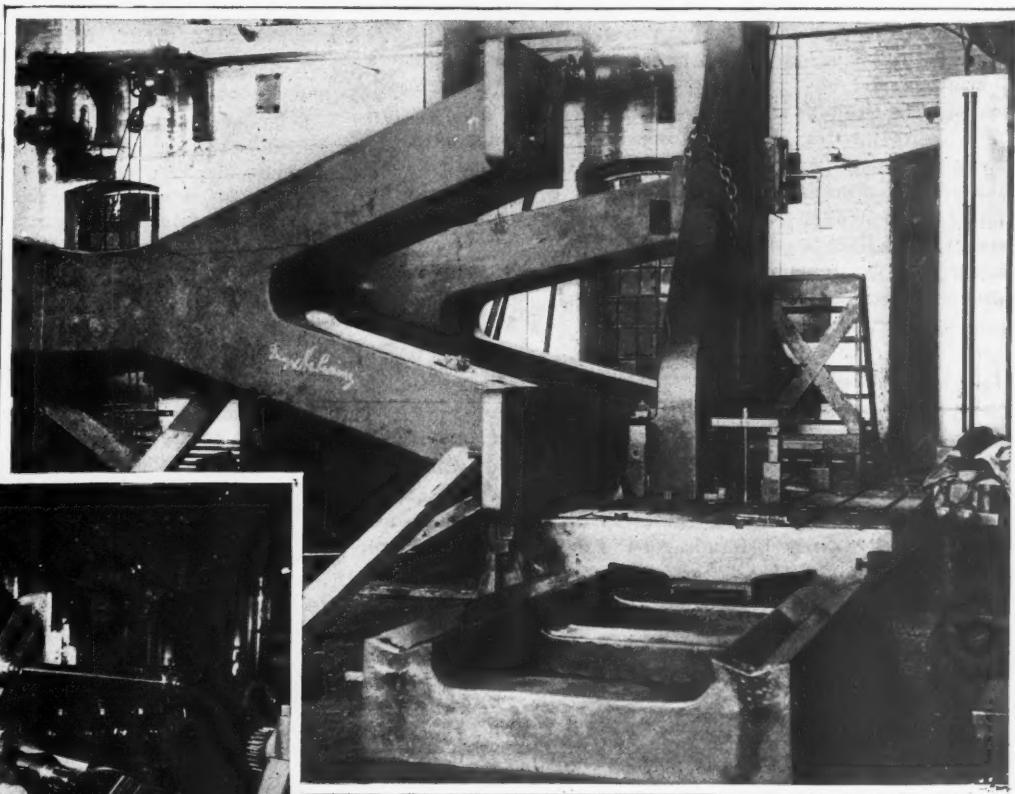


FIG. 5.—AN AWKWARD PLANER JOB.

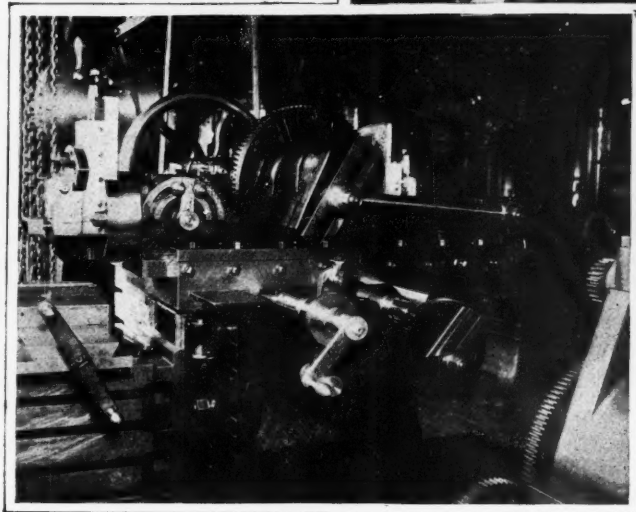


FIG. 6.—A MODERN SHAPER.—SEE PAGE 360.

fession in general is not duly appreciated, whereas there is more original brain work required in its practice than in almost any other branch of business. In many of the other professions the work is largely that of adapting and remembering book lore. While I do not mean to decry the position of the lawyer—far from it, for he belongs to one of the greatest of the professions—still, a little fling, so to speak, I think is not amiss. One of the principal requirements of a good practitioner is that of being well versed in the books of law, both ancient and modern, and while their brains and abilities are quite thoroughly taxed at times, in proving to a jury that black is white, or that a man "did not steal an article," but "simply appropriated it," there is not the need of original research and invention that there is in the engineering and scientific world.

There is a tendency among our engineers to keep the results of their research and practice to themselves, and it is not hard to understand the wish to hold what has often been obtained only after bitter experience, still it does not benefit them to any extent, and it would probably be of great advantage to the world at large. On the other side of the water particularly, there

tice. About this there is a wide divergence of opinion, some favor the extensive use of the note-book, while others approve of its abandonment almost altogether, unless for some special purpose as estimates, etc. As in all things there are extremes, and probably our best course is to choose a mean. It is a question whether the keeping of minutely dimensioned notes is a good thing, for we are apt to follow the figures rather than the spirit of the design, but the keeping records of the ideas themselves when special or unusual, is certainly a good scheme.

Owing to physical restrictions a man cannot carry unlimited notes in his mind, and should therefore aid his memory, where possible, with sketches and jottings on the subjects relating to his work, but the keeping of too promiscuous notes prevents a man using his ability to get up improved designs, and makes him more and more of an adapter and less of an originator. The extent and manner of keeping them must therefore be a matter for each to decide for himself. As regards the right of taking these notes, this must also be decided by each one. Frequently data, etc., about certain things cannot conscientiously be taken, and our better nature must consider carefully the subject, when there is any room for doubt as to the moral right. There are numerous handbooks on the different branches of the engineering science, but as a usual thing to use them requires a pretty thorough knowledge of the subject, for while the general formulæ are correct, modifications are necessary for each case, and they do not treat of the minor details of construction, where errors are most likely to creep in and on which depend the good

working of the machine. There is really a great need of reference books treating these subjects in a less theoretical manner. This need also makes the keeping of notes of great importance when kept in a practical way.

A good mechanic does not necessarily make a good draughtsman, and in fact generally, we might almost say, does not. The qualifications for each are different in many ways. A man may thoroughly understand the manner of making and finishing a piece of work, but when it comes to the planning of it originally, be all at sea. To become a competent designer requires much study and the beginning at the bottom of the ladder when young, so that the mind may be moulded to suit that broader field of the originating and adapting of ideas, such that the finished product will be as nearly perfect as possible, both in action and appearance.

Draughting is certainly a stepping-stone to something better, and it is but rarely we see a man grow old in the business. Take almost any draughting office, and you will find but a very small percentage of the men whose ages are over thirty-five, and even this is a high figure, and the usual run is between twenty and thirty. Where they all go to is almost a mystery, and probably an investigation would give quite interesting results. Of course some become chief draughtsmen, others engineers, superintendents, etc., but even these positions would hardly seem sufficient to take care of the supply.

The hours for draughtsmen are usually too long, as it is a wearing business, particularly so on account of the small amount of exercise. In the large cities the hours are usually good, but outside in the smaller places they are required to serve longer. Seven and a half or eight hours at the most should compose a day's work. Draughtsmen as a class are gradually being treated more leniently than formerly, and let us hope that the future will show even better results.

\* \* \*

## THE OUTLOOK FOR AMERICAN MACHINERY IN GREAT BRITAIN.

The foreign market for American machinery is of growing importance to manufacturers in this country, and they will doubtless be interested in the following summary of recent inquiries on this subject among a number of representative dealers and manufacturers in Great Britain. As the demand there for tools in 1896 was largely caused by the increased manufacture of bicycles, we have also sought opinions on the future of that industry.

Messrs. John Lang & Sons, lathe manufacturers and special machine tool makers, Johnstone, Scotland, say:

As far as we can judge, if the trade in this country is very good, then there will be a demand for American machinery. On the other hand, if trade is bad, there will be no demand.

As one who has visited a very large number of the American workshops, we do not think that it is possible to produce machinery as cheaply in America as in this country, providing that we have the same facilities in machine tools.

In the past, manufacturers in this country have been slow to adopt modern methods, but recently a very great improvement has taken place, so that now manufacturers here are very well equipped, and we are inclined to think that there will only be a demand for American machinery when trade is very brisk and when the manufacturers here cannot supply the demand.

As to the manufacture of bicycles and motor cars in Great Britain, we are inclined to think that this trade has not yet reached its maximum, and that it will go on for some time to come.

Messrs. Chas. Neat & Co., machine tool dealers, 112 Queen Victoria street, London, say:

As you are aware, for some very considerable time past, export of American machinery into this country has been on a very large scale, the cycle trade being undoubtedly the reason why so much has been called; for your manufacturers supplying to this market machinery more suitable than can be made here, and which for some considerable time seems to have been greatly appreciated. We have been importing machinery for some years past and have never known the quantity imported to have reached the value which it has been done of late. There will always be considerable demand in this country for American machinery, but the sales of the last eighteen months cannot possibly be maintained. There is no doubt that if your manufacturers continue to supply the class of machinery which they have been doing, no one in this country will be able to compete with them in the quality of the tools, and no doubt the demand will be at all times of a character sufficient to encourage your manufacturers to hold this market. The price at which the machinery is supplied is considerably lower than the same quality can be made here.

Respecting the manufacture of bicycles in this country, we think the

maximum has been reached, and that all the manufacturers have now complete plants and are not likely to increase to any great extent in the future. Small manufacturers have been running into the boom to such an extent that the supply now exceeds the demand, and we do not think that unless some extraordinary development should occur, or some new pattern should come out, the manufacture in the future will be to the extent that it has been. The year 1896 was a most unusual one, and we do not think the sales will ever again be reached in amount, although we should only be too pleased if they were. We think manufacturers have purchased so liberally that for some time to come there will be a great falling off in the machinery coming, which will be rather disappointing to your manufacturers, but there is no doubt a certain trade will be maintained.

Messrs. Webster & Bennett, engineers and machine tool makers, Coventry, say:

In our opinion that American machinery of high grade and special kinds will still continue to have a sale here, but ordinary standard machine tools such as screw-cutting lathes, milling machines and the general cycle machinery will not be sold in anything like the quantity that has been the case during the past two years.

There is every indication that the cycle boom has reached the maximum, and that English tool builders will be able to supply ordinary requirements in the near future.

Messrs. John Birch & Co., Ltd., merchants and engineers, 10 and 11 Queen Street Place, London, E. C., say:

In our opinion the sales of American machinery in England last year were due to two or three special causes, viz.: The unprecedented demand for bicycles; the general trade activity of English machinists, and complementary to this, the slackness of trade in America last year. Supplementary to these no doubt the excellence and adaptability of the American tools took buyers by storm. But the lesson is one not likely to be left long unlearned, for the British manufacturer, like most of his countrymen, is tenacious if conservative.

The prosperity and activity of last year still continue, and in the opinion of many will last for some time yet; though the falling off in a large market such as that of India, through famine, pestilence and other misfortune, is one to be counted with.

The future of the British market for American machinery is a difficult one. We are not of those who seek to draw comparisons, favorable or unfavorable, between American engineering methods and those of this country; we think that each is suited to its own environment. Notwithstanding this, we emphatically think that while we are ready to accept things of approved excellence from you, our manufacturers will not be slow to turn out the same description of article, or one with modifications to suit our market. You are probably ready to do the same, and we know that other countries are; we have therefore to accept the general rule.

As to the manufacture of bicycles reaching its maximum in this country, we think that it will this year, if it has not already done so. It is rash to prophesy, but there are signs that the trade is being over-done, and it will probably be for some years to come that production will not increase.

Messrs. Henry Kelley & Co., 26 Pall Mall, Manchester, say:

We are convinced that American machinery of certain kinds will continue to be sold in Great Britain in sufficient quantities to warrant a continued effort to keep this market. We refer to the best types of labor-saving tools you send over. So far as the medium and cheap types of standard tools are concerned, we are of the opinion they will experience a gradual but continued falling off.

We do not think the manufacture of bicycles has reached the maximum, for undoubtedly there will be quite a number of new factories started up for the manufacture of cycles; but you will not get anything like the same proportion of the orders that were given out last year.

To those who devote their energies to really first class tools, we would say, you have at least as good a prospect as ever you had.

Messrs. C. W. Burton, Griffiths & Co., dealers in machinery and tools, 158 Queen Victoria street, London, E. C., say:

We are of the opinion that American machinery of various kinds will continue to be sold in Great Britain for some time to come, and we do not think that there is a great probability of these tools being made in the near future.

The American lathe has been more or less on the market here for a number of years, but has practically never been copied, and seeing that the British manufacturers do not bind themselves to one article, but generally manufacture a varied number and styles of tools, we think they would be unable to compete in price with those made by specialists on your side.

We are of the opinion that the manufacture of bicycles has reached its maximum for some time to come, and that manufacturers are fully equipped for their present wants, and that the chief trade will be in special machines for special purposes, which show considerable reduction in the cost of any particular cycle part.

We do not think that this year will show a very large falling off in

the sales of American tools, but we do not think it will be larger than '96. As far as we personally are concerned we are considerably in advance of last year.

A widely known firm of Scotch engineers says:

We are certainly of the opinion that the new grade of high class machine tools, as lately specified and illustrated in the American tool catalogues, will continue to hold a first place in Great Britain. Our engineers now have a more favorable opinion of American made tools than they had a few years back. No doubt this very fact will stimulate our tool makers to further exertions both in design and finish, and they ought to produce at less cost than you import. Since early in '96 all our tool makers have been overwhelmed with orders; this no doubt has increased the demand for American tools, and it is our opinion that with continued effort and a more widespread representation than hitherto, there is no valid reason why they should not keep the hold now gained.

Whether the boom in cycles has reached its height or not it is difficult to forecast; but we are quite satisfied the British makers are fully equipped and more than able to meet all future demands.

The following opinions represent three well-known London firms, who, on account of trade connections, prefer not to have their names used. The first is one of the largest dealers in Great Britain, the second is both a dealer and a manufacturer, the third is a manufacturer:

(1) While English machine shop practice has advanced, it has not done so in anything like the proportion that the American goods have done, and so far from diminution in regular trade, we are certain that for years to come there will be a steady increase in England in regular machine shop tools. We have no fear for some time to come of the English makers competing with similar tools, because it will be such a revolution of their practice here that very few firms would undertake it. Where the English build one tool, the Americans build twenty or even one hundred of the same class. Take milling machines as an example: we have no doubt that trade reached high water mark in 1896, but we must confess that we have not found very much falling off in orders.

As far as manufacturing bicycles goes, most of the fac-

sales to be made to the classes who cannot afford to take up the cycle at the present price.

With due allowance for the fact that these opinions are given by firms who are more interested in seeing home trade flourish than in having American tools brought into their market, the net result of the inquiry indicates that a continuation of the unceasing effort for improvement which characterizes the mechanics of this country and which has enabled them to enter foreign markets, will, in certain lines at least, keep them in advance of their foreign competitors, so that a market for American machinery will continue to exist in Great Britain, unless the cost of labor and material in this country should materially advance, which conditions would necessarily be accompanied by a revival in our home market that would more than compensate for the resulting loss of foreign trade.

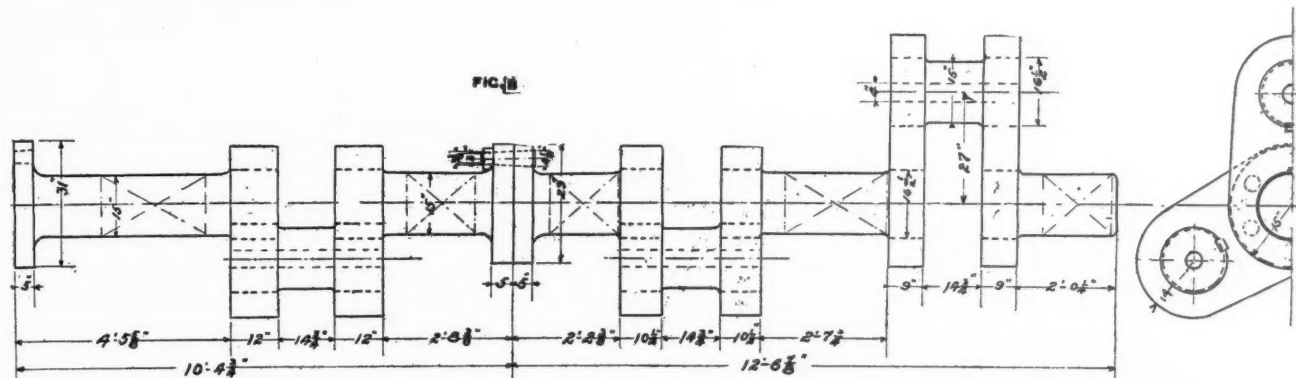
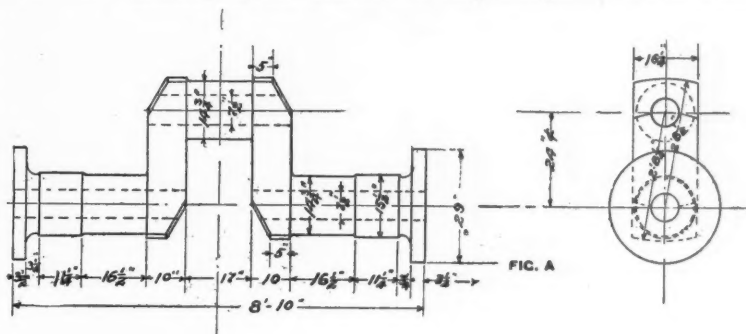
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## MARINE ENGINE DESIGNS.\*—2.

WILLIAM BURLINGHAM.

### CRANK AND LINE SHAFTS.

The dimensions of crank shafts, as obtained from the formulas in this article, are practically correct and in accordance with present marine practice. There are many points in connection with the turning and bending moments that have not been considered, for the reason that they complicate the formulas ex-



ories are now so fully equipped, that with the number of machines they will be able to turn out in the near future there will not be much demand for more machinery, but while the bicycle section has been good during the past two years, it only accounts for a small proportion of the English machine tools, and it is the wakening up of the old fashioned ideas in other quarters that we look forward to in the future for a still more prosperous time than in the past.

One of the causes of success in American machine tools in this country is owing to the promptness with which they have been delivered, and not because the English maker was unable to compete. Wherever you go in England you will find manufacturers busy running full time, and four, five and six months behindhand with orders, so there is evidently a good prospect for all for some time to come.

(2) It is somewhat doubtful as to whether American machinery manufacturers will, in the majority of cases, be able to profitably export the general run of ordinary machine tools; but manufacturers of good specialties will probably be able to hold their own.

With regard to the manufacture of bicycles, of course every day there are more small manufacturers of cheap machines springing up, and it is very hard to say, just at present, whether this trade has reached its maximum, as of course in this present year so much much money is being spent on other things, which of necessity prevents many persons from investing in cycles.

We think it probable that British manufacturers will do their utmost to cut the prices down in the cheaper class of goods all around.

(3) We think the cycle trade will hold its own for several years, as the result of competition between English and American manufacturers must be a reduction in price, which again on its part will cause more

ceedingly and have but little effect upon the results.

A few years ago crank shafts were invariably solid, that is in one piece, but with the advent of triple and quadruple engines, they were made in two parts for the merchant marine, and in three or four parts, reversible and interchangeable, for the navy. When made in two pieces they are generally cut between the mean and low pressure cylinders. The modern cranks are nearly all of the built-up type, the crank webs being shrunk over the shaft and pins. This insures a more reliable shaft, as the forgings are smaller and the holes through the webs allow of a better examination of the web metal. Large shafts are now forged with an axial hole through their entire length, thereby reducing the weight with a comparatively small reduction of strength, and insuring a shaft of uniform material and quality.

In general the process of making hollow steel shafts consists of casting an ingot of twice the diameter required when finished and 25 per cent. longer than necessary, as this amount is cut from the upper end to make sure that there is no dross in the finished product. The ingot is then bored approximately to the required diameter of the axial hole, and, after reheating, a mandrel is inserted and hydraulic pressure applied until it is drawn out to the required sizes. The reheating should be done with great care to allow for a slow and uniform penetration of the mass.

\* In this article I am greatly indebted to Mr. G. Edward Smith for his assistance.

Annealing is done by heating the steel slowly in a furnace and allowing both to cool together. Tempering is effected, first, by heating the forging to a temperature varying with the use to which the forging is to be put, and then plunging it into a bath of oil. It is then carefully annealed. The advantages derived are hardening and a breaking up of the crystalline structure due to forging and a modification of the physical properties, increasing the elastic limit and adding toughness.

In determining the size of the crank shaft we will consider only the twisting moments, that is, the unbalanced load, in pounds, on the piston multiplied by the distance in inches of the point of application of this load from the center of the shaft, and the bending moment on the crank pin next the propeller end of the shaft. The bending moment to be used is the product of the unbalanced load on the piston by the distance of the center of the pin from the center of bearing divided by four. The calculation of the effect of the momentum of the moving parts is rather difficult for practical purposes and it is better to allow for it in the factor of safety. As an ordinary crank shaft is subjected to both twisting and bending simultaneously, the resulting strain

For practical purposes the load may be considered equal in each of the three cylinders.

Assuming 115 lbs. as the max. unbalanced load per sq. in. on the H.P. piston and multiplying by the area of the H.P. piston, we have a total load of 101 500 lbs.

unbalanced load  $\times$  dist. between bearings.

$$mf = \frac{101\,500 \times 37}{4}$$

$$mt = \text{crank radius} \times \text{unbalanced load} \times \text{constant.}$$

$$mf = \frac{101\,500 \times 37}{4} = 938\,875.$$

$$mt = 24 \times 101\,500 \times 2 = 487\,200.$$

$$\text{M. E. T.} = mf + \sqrt{mf^2 + m^2 t^2}.$$

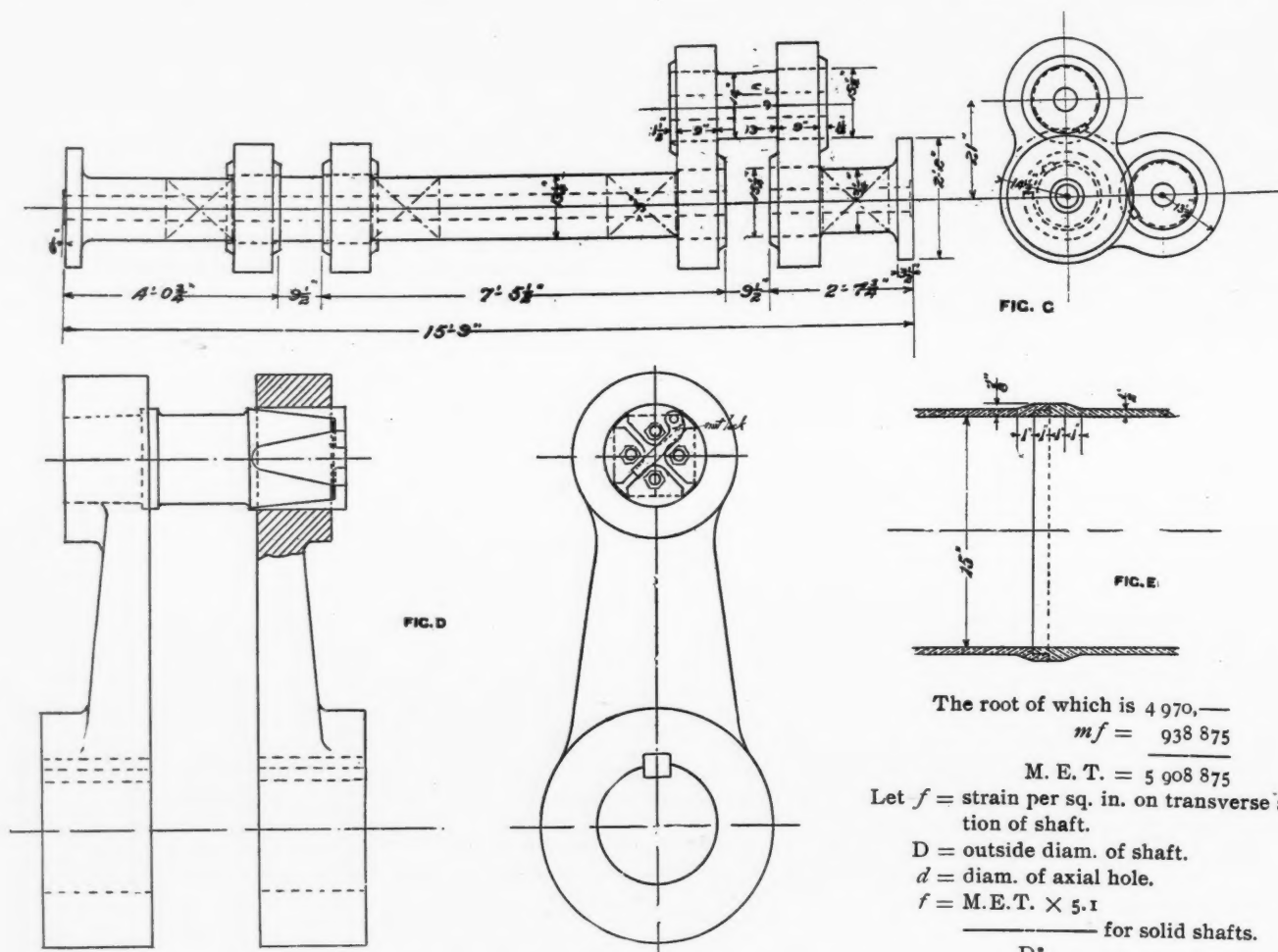
$$\text{M. E. T.} = 938\,875 + \sqrt{938\,875^2 + 487\,200^2}.$$

Sufficiently close results may be obtained by the use of the slide rule, as follows:—

$$mf^2 = 881, \text{---,---,---}$$

$$mt^2 = 23\,800, \text{---,---,---}$$

$$24\,681, \text{---,---,---}$$



on any section may be measured by reducing these two strains to one known as the equivalent twisting moment.

In a simple engine the twisting moment is that due to the load on a single piston, but in a compound, triple and quadruple the moment is a combined twisting moment due to the loads on the several cylinders. This combined moment is not equal to the sum of the loads of the respective cylinders multiplied by the throw of the crank, but varies with the changes of the crank position; the maximum combined twisting moment is the only one that need be considered.

For example, we will figure the crank shaft of a vertical, inverted, direct acting, triple expansion engine  $\frac{33\frac{1}{2}}{48}$  and 51 and 78.

Steam pressure 195 lbs. absolute and 120 R. P. M. to develop about 5 000 I. H. P. Fig. A shows one part of the shaft. It is interchangeable and reversible and is made of forged steel:

Let Bending moment =  $mf$ .

Twisting " =  $mt$ .

Maximum equivalent twisting moment = M. E. T.

The root of which is 4 970,---

$$mf = 938\,875$$

$$\text{M. E. T.} = 5\,908\,875$$

Let  $f$  = strain per sq. in. on transverse section of shaft.

$D$  = outside diam. of shaft.

$d$  = diam. of axial hole.

$$f = \text{M. E. T.} \times 5.1$$

for solid shafts.

$$D^3$$

$$f = \frac{\text{M. E. T.} \times 5.1}{\frac{D^4 - d^4}{D}} \text{ for hollow shafts.}$$

Substituting in the last equation we have

$$f = \frac{5\,908\,875 \times 5.1}{(47\,400 - 3\,164) \div 14.75} = 10\,050 \text{ lbs. per square inch.}$$

This stress should be from 8 000 to 10 000 pounds per square inch. The pins and shafts should be made of the same diameter, the length of the pins to be about equal to their diameter, preferably a little less, the pressure per square inch being from 275 to 315 pounds for merchant work and about 400 pounds for naval work. The journals should have about the same pressure assuming that the entire load of piston is on one journal.

A short and reliable formula for solid steel shafts is the following:

$$D = \text{diameter} = \text{constant} = \sqrt[3]{\frac{\text{I. H. P.}}{\text{Rev. per min.}}}$$

The equivalent diameter of a hollow shaft,  $D$ , is

$$D_1 = \sqrt[3]{\frac{D^3}{1-x^4}}$$

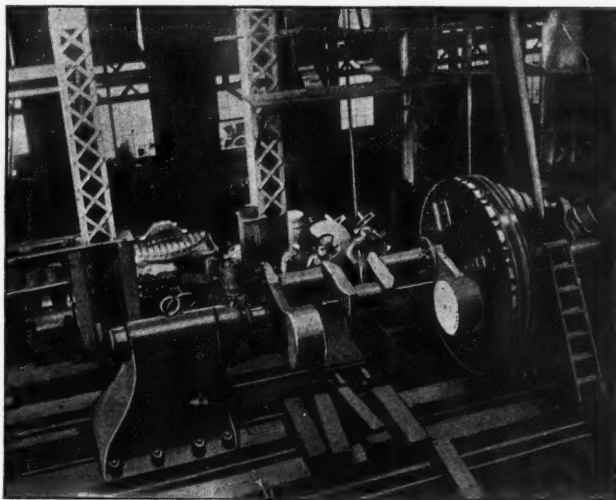
where  $x$  is the ratio between the diameter of the axial hole and that of the shaft. Using the same example as before with the constant = 4.15 we have

$$D = 4.15 \sqrt[3]{\frac{5000}{120}} = 14.37$$

and with  $x = .508$ ,

$$D_1 = \sqrt[3]{\frac{2975}{1-.0666}} = 14.72$$

the outside diameter.



READY FOR TURNING.

The following is a table of constants compiled from lately built vessels with steel shafts:

1 000 ton gunboat.....	4.3	5 300 ton vessel.....	4.24
2 000 " " .....	4.1	7 500 " " .....	4.3
3 000 " " .....	4.3	11 000 " " .....	4.15

This gives an average of 4.23, which average constant is correct for everything except very heavy shafts of good material; these latter could have a constant of from 4.15 to 4.2. The constant for wrought iron shafts is 4.55.

The American Shipmasters' Association require the shafts to be of not less diameter than is determined by the formulæ hereafter set forth:

"The tensile strength of cast steel shafts not to exceed 68 000 pounds per square inch, and a test piece cut from shaft should bend cold, without crack or fracture, to an angle of 100 degrees over a radius not exceeding two inches.

"Forged steel shafts to be of not less tensile strength than 58 000 or more than 68 000 pounds per square inch of section; the test piece, which must be cut cold from the shaft forging, to show an elongation of not less than 20 per cent., if their length is 8 inches, or 25 per cent. if their length is 4 inches.

"All shafts must be carefully examined after being turned, preparatory to being fitted."

$d$  = diameter of shaft in inches.

$P$  = absolute pressure in pounds per square inch.

$S$  = stroke in inches.

$D$  = diameter of low pressure cylinder in inches.

$C$  = constant according to the following table:

Style of Engine.	For Crank and Propeller shafts.	For Intermediate shafts.
Double expansion.....	10 286	11 714
Triple " .....	17 905	20 190
Quadruple " .....	19 905	22 388

$$d = \sqrt[3]{\frac{P S D^2}{C}}$$

All ships desirous of a rating in this association must have their shafts at least equal in diameter to that given by these formulæ.

The breadth of the crank arm of a solid shaft is 1.1 times the diameter of the shaft; this allows a good fillet around the pin and shaft. The thickness of the arm is from .68 to .7 of the diameter of the shaft in naval practice, and about .75 of the diameter in the merchant marine. Solid shaft crank webs are usually made the same thickness for all cranks.

In the merchant marine, crank shafts are generally built up (see Figs. B and C). Great care is required to properly construct such a shaft so as to have it perfectly true when properly finished and to have the arms shrunk on without leaving the metal around the pins and shaft-ends in such a state of tension as to be dangerous. Shafts above 12 inches in diameter are better if built up, though smaller shafts are often made thus, as they are deemed less expensive than the solid ones. Built up shafts are not used in naval work, on account of the saving of weight possible by the use of solid shafts. In ordinary marine work it is customary to increase the thickness of crank arms for each cylinder, directly proportional to the load they have to carry. The diameter of the crank web, at the shaft should be twice the diameter of the shaft, and at the pin 1.9 times that diameter. Of course these dimensions are variable, but the average of a number of good engines will be found to give these ratios.

Couplings are nearly always forged solid with the shaft, and in thickness should be from .25 to .28 of the diameter of crank shaft; the diameter is fixed by the number and diameter of the coupling bolts. The pitch circle should be of such a diameter as to allow the use of a drilling and reaming machine alongside the shaft. On large shafts at least three inches should be allowed from the outside of the bolt hole to the outer diameter of coupling when the taper bolts are used, as with less, the metal is very apt to bulge around the outside diameter of the coupling.

Coupling bolts have practically only a shearing stress to resist, consequently their size is determined by the following formulas:

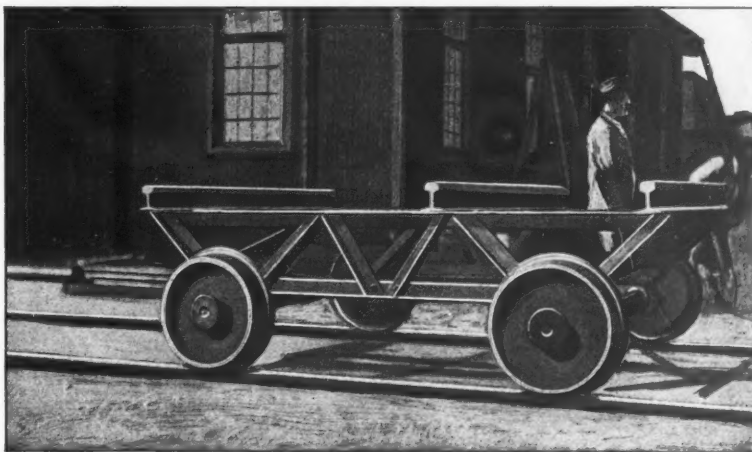
$$\text{Area of one bolt} = \frac{.196 D^3}{N \times r}$$

for solid shafts, or

$$.196 \left( \frac{D^4 - d^4}{D} \right) \\ N \times r$$

for hollow shafts, where  $N$  = number of bolts and  $r$  = radius of pitch circle.

The screwed part of large bolts may be reduced  $\frac{1}{4}$  to  $\frac{3}{4}$  inch in diameter, and the thickness of the nut to about .75 of diameter of the threaded portion. The taper of coupling bolts should be about one quarter of an inch to a foot on the diameter.



CAR FOR HANDLING THE HOT FORGINGS.

The crank pins of a paddle wheel engine are subjected to slightly different stresses from the foregoing. These are usually of the intermediate shaft type, in which the outer crank arms are keyed upon the paddle shafts, while the inner ones are similarly fixed to an intermediate shaft, the crank pins being fixed in the inner crank arms and left, to a certain extent, free in the outer ones, as shown in Fig. D.

Strain per square inch =

$$\sqrt[3]{\frac{\text{Max. unbalanced pressure} \times \text{distance between webs} \times 10.71}{(\text{shaft dia.})^3}}$$

Steel pins in these cases are allowed about 7 500 pounds per

square inch of section. The pressure on bearing surface of pin should not exceed 700 pounds in compound engines, although the pressure on crank pins of some very good simple paddle-wheel engines runs as high as 1 400 pounds per square inch.

The keys for cranks should be made of steel, somewhat harder than the shaft and crank; it is better to figure the dimensions of all keys used in engine work rather than to use the published

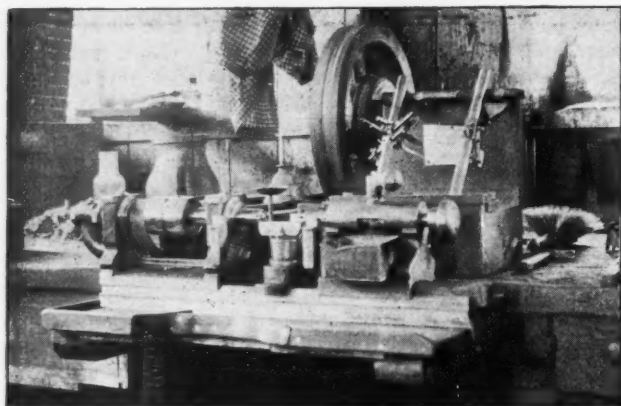


FIG. 1.

tables of line shaft keys. We should consider the keys firmly fixed and figure them for direct shear. The method of fixing the loose ends of the crank pins, as shown in Fig. D, is one of many ways, nearly every firm having its own particular method.

The tail shafts of screw engines must be strong enough to resist the bending stresses arising from the weight of the propeller, and its action when wholly or partially submerged, also the twisting stress due to the engine. Experience has shown that the tail shaft should have a diameter not less than that of the crank shaft, and it is often made from 5 to 10 per cent. greater than this. The reason for this is that the shaft is so inaccessible and difficult to replace, and its breakage is very apt to cause the loss of the ship.

The taper for that portion of the shaft to which the propeller is attached should be one inch in a foot on the diameter.

The intermediate shafting should be about  $\frac{1}{10}$  of the diameter of the crank shaft, as it has to transmit the twisting strains only. Drawings to be sent away to the forge shop are better made to finished sizes and marked as follows:

Finished sizes are marked.

To be rough turned to within  $\frac{1}{16}$  inch of sizes marked, thus allowing  $\frac{1}{16}$  inch finish all over (for medium and large shafts).

The crank pins must be accurately parallel to main journals.

All work to be straight and turned true.

Allow finish as specified in all shafts.

Each length of shafting to be forged in one piece.

The photograph shows a very good method of balancing a section of a crank shaft in a lathe. It is advisable to balance these in some similar fashion both for the sake of the lathe and to enable a smooth, even cut to be taken.

In shrinking a pin or shaft it is customary to allow about two papers, in shop parlance, or about .006 of an inch. These pins and webs can be very effectively handled, while hot, on a car like that shown in the second photograph.

The tail shaft is usually covered with a composition sleeve, varying in thickness from  $\frac{7}{8}$  to 1 inch, according to size of shaft, the Government formula for which is, copper, 88 per cent.; tin, 10 per cent., and zinc, 2 per cent., although nearly every shop has a different one.

These sleeves are, at present, shrunk and pinned on, and for shafts above 12 inches diameter should be not more than 4 feet 6 inches long. The difficulty of shrinking them on, increases very rapidly as they exceed this length. A very good joint is shown in Fig. E. All these joints should be absolutely water-tight, and the sleeve should make a water-tight joint with the propeller hub.

Fig. B is the crank shaft of a  $\frac{28-44-74}{54}$  vertical inverted triple expansion engine with 180 pounds steam pressure. Fig. C, one piece of a shaft for a  $\frac{24-34-48-68}{42}$  vertical inverted quadruple expansion engine, 210 pounds steam, shaft in two pieces interchangeable, but not reversible. The webs are cast steel and the bosses on the webs around pins and shafts are introduced for the purpose of allowing the engine to be made shorter. This is not advisable, as it is dangerous for the engineer to feel the crank pin with this style of web. \* \* \*

## NOTES IN A CLOCK FACTORY.

FRED H. COLVIN.

A very interesting combination tool, or perhaps it might more correctly be called a gage tool, is shown in the illustration, Fig. 1. Two of them are shown leaning against the tool box, and consist of a round bar of steel with numerous cutters inserted at various points, the proper distances apart. In use the work, which is mostly spindles used in the clock works, is held between centers of the small bench lathe and the bar laid across the rests in front. Then by swinging the tool into the proper position and moving as required, the spindles are turned in the proper places, with shoulders in the correct location. This idea is capable of application in other places and should prove interesting.

The wooden case department of this plant, the E. Ingraham Co., of Bristol, Conn., is very well equipped modern tools and appliances. The lower view of group shows a sand-papering disc which is very useful, and not as well known as its importance and convenience deserves. It is about four feet in diameter, and, as shown, the covering of sand-paper is put on in halves. The shield or hood is connected with the exhaust system for

removing dust, and the men suffer no inconvenience from this at any of the machines, which is an important item in wood-working shops. The upper view shows the main exhaust pipes on their way to the fan, and also the manner in which the pulleys having belts running through the floor above are boxed to prevent said belts from dropping when thrown off the upper pulleys.

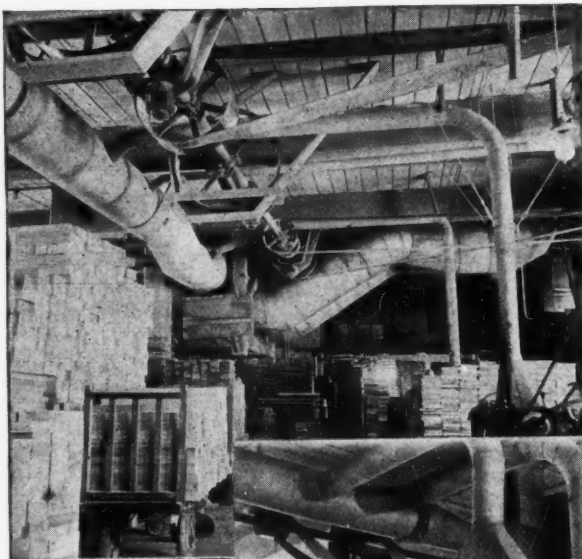


FIG. 3.



FIG. 2.

Those who are familiar with the appearance of pulleys in most wood-working shops, where the dust flies at will and lodges in pulleys and on shafts until they assume massive proportions, will see that here the arms and rims are as clean as can be desired, and this speaks well for the efficiency of the exhaust system in use.

## PRACTICAL PATTERN MAKING.—3.

I. McKIM CHASE

## THE MANAGEMENT OF A MODERN PATTERN SHOP.

The qualifications necessary for a foreman to possess in order to successfully manage the affairs of a large pattern shop are that he should be a draughtsman, a good arithmetician, should have a thorough knowledge of the art of moulding, and should be a good judge of human nature as well as of the different materials used in his department. He should be able to decide the manner in which any pattern is to be moulded, and to direct the construction of the pattern accordingly. He should also have a thorough knowledge of the construction and care of wood-working machinery, and not the least of the necessary qualifications are energy, good character and habits.

By some persons the pattern shop is considered a drawback to the machine business in consequence of the expense of making patterns; for they do not show in a completed structure as other materials do, and are considered unproductive. Yet the pattern shop is more essential than the drawing department, to which it is closely allied. It is possible to dispense with the latter in the machine business, though not with the former where castings are required. But unfavorable criticism of the pattern shop is frequently the result either of the critics' inexperience in mechanical pursuits, or the assumption of knowledge that he does not possess. The person who invents a method of making castings without the aid of patterns has both fame and fortune awaiting him.

The expense attendant on the use of patterns is often unnecessarily increased, owing to the abuse which they receive in the foundry. Some moulders are veritable pattern smashers, and will do more damage to a pattern in making a half dozen castings than others will in making a hundred.

Pattern work, like all other kinds of model work, is expensive, and can be made more or less so according to the work required of the pattern. In this respect patterns may be divided into three classes, and the cost of producing them should be varied accordingly.

a. Patterns of a temporary character, those not likely to be used more than once. These should be made with as little expenditure of labor and material as possible to enable them to perform their functions. These patterns should not be preserved, as they unnecessarily encumber the pattern loft.

b. A class of patterns likely to be used occasionally, sometimes at long intervals. These should be preserved, and more pains be taken in their construction than with the former, as they have to withstand the usage in the foundry as well as the distortion likely to occur to them during their storage in the pattern loft.

c. A class of patterns regarded as standard and which are frequently used. These cannot be made too well, and when properly constructed are necessarily expensive in first cost.

When a drawing is received in the pattern shop, the first duty of the foreman in connection therewith is to acquaint himself with it and decide how the pattern is to be made, and in what manner moulded. If detail drawings of a machine or other device to be constructed are received, a general drawing should accompany them, or else the foreman should be made acquainted with the general arrangement of the parts. When this is done, he will often be able to detect errors which might not otherwise be discovered until after the castings had been made and the machining of them in progress.

There are several allowances necessary to be determined previous to beginning the construction of a pattern. The one most troublesome to the pattern maker is that for finishing. The amount that will answer for one machinist will not suit another. It is advisable to leave as little as possible for finishing, and to have sufficient to allow for the proper finishing of the castings. This allowance will depend a great deal on the result of the casting and its likeness to the pattern. This is likely to vary according to the manner of moulding the pattern. As a rule, the castings requiring the greatest amounts for finishing are those which have been moulded in loam, and castings made of steel. These are liable to vary from the proper dimensions to a greater extent than those moulded by the other methods. Large castings of steel are never as true to pattern as those of other metals.

For patterns to be moulded in loam and for steel castings an

allowance of from one-fourth to one-half of an inch, according to the part to be finished, is necessary.

For ordinary castings moulded in green or dry sand an allowance of from one-eighth to one-quarter of an inch is sufficient. For the smaller castings, which have been moulded neatly and are of sound metal, an allowance of from one-sixteenth to one-eighth of an inch will answer.

The allowance for shrinkage, or the amount the pattern is required to be made larger than the intended casting, is another important preliminary matter to be determined before constructing a pattern. The conventional allowance for iron castings is one-eighth of an inch per foot, but this rule needs modification in applying it to castings of various shapes, dimensions and mixtures of metals. To insure accuracy in castings much depends on the judgment of the pattern maker in providing for their construction. Hard irons, as gun iron, will shrink more than the above amount, while soft iron will shrink less. Yellow brass will shrink more than bronze. A plain cylinder will shrink less in diameter than in length.

With large cylindrical or box shaped castings of iron, it is good practice to allow one-tenth of an inch per foot for shrinkage in length, and one-half of this amount in diameter, or across. The shrinkage in length of such castings is generally very little restricted, while in diameter it is resisted by the cores or internal parts of the mould. Two castings of the same weight and of the same kind of material, one of which is extended and the other more compact, will shrink differently, the latter shrinking less than the former.

Metals, like water, are densest in their liquid state, the point of greatest density being near the temperature at which they solidify. From this point they will expand either with a reduction or an elevation of temperature. Iron, when about to solidify, undergoes a sudden expansion, owing to the effort of the molecules to arrange themselves in definite positions. After solidification takes place it begins to contract, with a further loss of temperature. When the contraction begins, the metal is just leaving its plastic condition, and its cohesive strength is considerably below that of its normal state. If at this period the contraction of the metal is resisted by parts of the mould, a fracture of the metal is likely to occur. With some of the more contractible metals, as with steel, to avoid fractures it becomes necessary as soon as the metal has set, to relieve the interior parts of the mould and allow the metal freedom in shrinkage. In the case of a plain cylinder, where its shrinkage is resisted by an internal core, the metal will contract within its annular wall until its cohesive strength becomes sufficient to compress the core, at which period it will have undergone part of its contraction. This accounts for the reduced shrinkage of cylinders diametrically.

The usual allowances for the shrinkage of castings of different metals are, per foot:

For iron.....	one eighth of an inch.
" bronze.....	five-thirty-seconds of an inch.
" brass.....	three-sixteenths of an inch.
" yellow brass.....	seven thirty-seconds of an inch.
" steel.....	three-sixteenths of an inch.
" aluminum.....	seven thirty-seconds of an inch.
" zinc.....	seven thirty-seconds of an inch.
" lead.....	seven thirty-seconds of an inch.
" tin.....	three-sixteenths of an inch.

It is not always known where the castings for which a pattern is to be constructed are to be made. The opinions of moulders will differ widely as to the best method of moulding some patterns. In such cases the foreman is often perplexed. His desire should be to always have a pattern made to be moulded to the best advantage of the foundryman. Where there is any doubt as to the best way of moulding a pattern, the foreman moulder should be consulted, where it is possible. As he is responsible for the proper production of the castings, his desire should be regarded and the pattern made for his convenience.

It is too often the case that strained relations exist between the heads of the pattern shop and the foundry in consequence of the perversity of one or the other, or through attempts made to shift responsibilities. Each should desire harmony in their business intercourse, because without this the work cannot be carried on to the best advantage of their employers.

The foreman of a pattern or any other shop should be relieved of any clerical work. His proper place is in the shop among the workmen, observing what is going on; to inspect and direct the

work in progress; to see that every employee is performing his duty properly, and that the materials and machinery are properly used. When he performs all this, he will have little time to devote to office work. With pattern lumber at six to seven cents per foot, and where large quantities are being used, it is an important part of a foreman's duty to see that it is economically employed. The repairs to machines, belting, etc., and the sharpening of cutters, is quite an item in the running expenses, and the desire should be to reduce this to a minimum.

A foreman should have full control of the employees in his shop as long as he is held responsible for its management. Without this it is probable that by some he will not be respected as he should be. He should be gentlemanly in his intercourse without being too familiar with his subordinates, and should insist on being respected by them. In some instances the responsibility of employing and discharging employees, as well as other duties which should belong to the foreman, are assumed by others above him. Where such a condition prevails, the inevitable tendency is to impair the efficiency of the shop, and it behooves the foreman to use his judgment very discreetly if he desires to reduce to a minimum the annoyances inseparable from such a system.

One thing that reflects credit on the management of a pattern shop is to have it look clean and tidy. Of course it is impossible, where so many shavings, etc., are made, to have such a shop look as clean as some other kinds of shops. However, it can be kept reasonably clean without an excessive amount of labor by a proper system, making it the business of a person to clean the shop. What helps to make a pattern shop look untidy is the accumulation of scraps, etc., that litter the floor under the work benches, thrown there by the workmen for future use, but who seldom trouble themselves to look through the lot when it is easier to cut a board. This accumulation is aided by the unsuitableness of the ordinary carpenter's work bench, which is the kind usually supplied to pattern makers. With the style of bench previously illustrated and described, having under it a shelf about one foot from the floor for the reception of articles not wanted for immediate use, the space under the bench can be swept clean and accumulation of rubbish prevented.

It is too frequently the case that work benches are unnecessarily abused. Some workmen will use the bench stop while sawing and thereby risk cutting into the top and vise rather than take the trouble of making a bench hook. The undue disfigurement of a work-bench is infallible evidence that it has been occupied by a careless and sloven workman.

The machines in the pattern shop most likely to cause accidents as well as to be misused are the circular saw and hand planer. When workmen are careless or ignorant of the use of machines they should be instructed how to use them properly. No saw should ever be forced beyond its limit for doing good work. Even a good saw in the best of condition can be made to work unsatisfactorily by forcing the work too hard upon it. In using a circular saw, a person should never place his hand behind it while standing in front, nor even let the hand pass in front of the saw while so standing. A stick should be kept handy and when the end of the work is near the saw, finish by pushing it through with the stick. Should the saw incline to run out when not forcing it, withdraw the work and investigate the cause, which will likely be one of the following: a dull saw, or one with insufficient set. Should the work spring and bind on the saw, withdraw it at once and begin sawing at the other end, or else have someone insert a wedge after the end has passed the saw. Many deaths have been caused by the board being sawn, binding on the back of the saw, which causes the board to be raised until the top of the saw comes in contact with it and throws it forward with great force.

The band saw is not considered a dangerous tool, but it is liable to great abuse by the use of saws that are too dull or insufficiently set, or by attempting to saw curves smaller than those in which the saw will freely turn.

Nearly every accident occurring on the hand planer is caused by attempting to plane short pieces which, before they are made to bridge the mouth of the planer, are caught by the knives and drawn in. Often a hand goes in with the piece of work, and the person is maimed for life. A good rule to be observed in using this machine is never to attempt to plane a piece of work on it less than ten inches long, nor less than  $\frac{5}{8}$  of an inch thick.

## MAKING A PROPELLER WHEEL PATTERN.

GEO. W. DEAN.

In making the pattern for a propeller wheel, some pattern-makers use but one templet, while others prefer to use more, especially in building the pattern for an irregular pitch blade. These templates are made of sheet iron cut to represent the different angles and widths of the blade, as shown in the draft from which it is to be made. A center mark is made on each templet, after which they are fastened to a bench or table in position, according to the different angles which they represent (see illustration A). When placed in their relative positions (if the wheel is to be a true or regular pitch) the center line will run horizontally from the center of the wheel's hub, through the center points marked on the templates.



METAL TEMPLATES (A).

With the exception of a few shorter pieces used in building the sides of the blade, the lumber selected for the pattern is cut in lengths equal to half the diameter of the wheel. It is planed to a uniform thickness; some preferring it to be an inch and a half and others two inches thick.

The different widths of the plank are obtained from the draft at the sectional views, as each piece varies in width, as shown in Fig. 1. One of the shorter pieces of plank is selected, placed on the bench and slid up against the templates. As the bevel on each templet is just equal in length to the width of the blade at the different sections at which they are placed, it will be observed that this first plank will only come in contact with the templates that represent the widest part of the blade. The edge

of the plank being square and the templet being oblique, only the lower edge of the plank will come in contact with the templet. The plank is marked at the points of contact and cut out about one-half inch each side of the mark, on a bevel with the templates. Each of these bevels must be cut so that when the plank is placed up

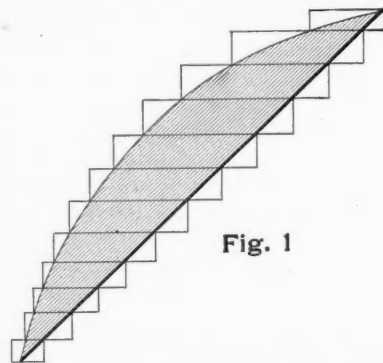


Fig. 1

against the templates they will fit accurately. The second plank must be a little longer and also wider than the first one used. This is placed on top of the first and fitted to the templates in the same manner. After fitting three or four, each additional one being a little longer and wider than the one which precedes it, they are firmly glued together and held in place by hand-screws and dogs until dry (see illustration B). To make them more secure they are afterwards doweled. The pattern is built in the same manner until the plank strikes the lower edge of the shortest templet, and also continues to the hub.

There are several methods of building the hub, three of which I will mention. One is to build up the hub and turn it to the desired size and shape, using lumber the same thickness as that used in building the blade. In building this way one side is left

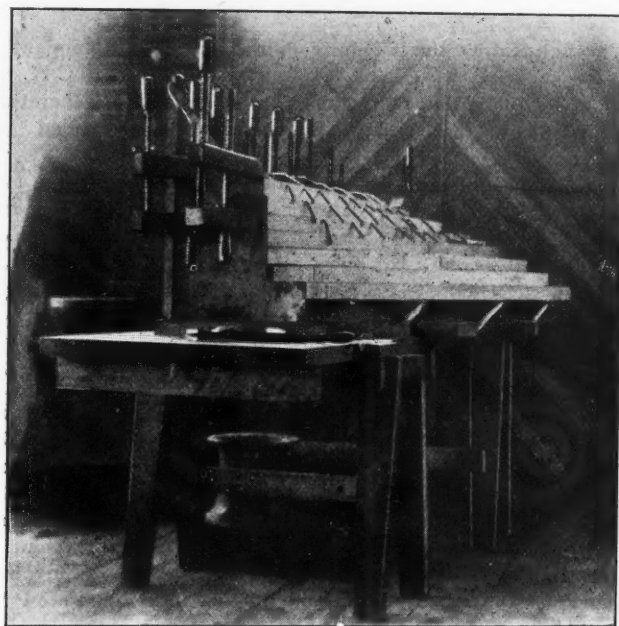
open in such a shape that in laying up the blade the planks will fit into the places left to receive them.

Another way is to let each plank extend to the center of the hub, leaving it wide enough to form one-quarter of the hub, which is all that is necessary.

Still another way is to build up the blade separately and attach the part of the hub with lag-screws. The illustrations represent a wheel built in the manner last mentioned.

In continuing to build the pattern each plank is extended to the hub and glued firmly until the blade is built up to the upper points of each templet. Pencil marks are made on the pattern at the upper and lower points of the templets and also at the center marks, so that the width of the wheel at different sections can be ascertained later. The pattern is lifted from the table and turned over. All the wood is cut off between the grooves (made by fitting the planks to the templets, and smoothed off to regular form from one groove to the other. A line is drawn from the center of the hub through the center marks to the end of the blade. If this be a true or regular pitch wheel, a straight edge can be used in drawing this line. If an irregular pitch, it would be necessary to use a flexible piece of wood or iron, as the center marks would not be in a straight line.

One-quarter or five-sixteenth inch holes (about three inches apart) are bored into the pattern along the center line. Dowel pins are driven into these holes and glued. They must be the exact length that the pattern is required in thickness. For



THE PATTERN GLUED UP (B).

example, if the section nearest the hub is five inches in thickness on the center line, the pin should be just five inches long. If the next section is four inches in thickness, the pin should be just four inches long. The pins between these sections should gradually decrease in length from five to four. Pins are also driven at right angles to these center line pins at the different sections, as many being used as the pattern-maker desires, as they are to determine the thickness of the pattern.

The width of the blade is traced out from marks that were made on the pattern at the upper and lower points of the templets, and from referring to the drawing. The edges of the pattern are trimmed down to these marks. The pattern is turned over and this side trimmed off to the end of the pins in regular form from one pin to the other. Generally a large fillet is put in at the hub of a size to get the desired form. A hole is bored through the hub the size of the spindle that is used in moulding it. The pattern is smoothed and shellaced, all the necessary finishing done, after which it is ready for the moulders.

\* \* \*

"You can't make a silk purse out of a sow's ear," was doubtless a wise and a very pertinent saying of some ancient philosopher. Wouldn't it be just as pertinent if another was coined to the effect that you can't make a sow's ear out of a silk purse? It makes all the difference in the world how you are looking at a thing.

## ALUMINUM IN KNITTING MACHINERY.

GEORGE D. RICE.

Five years ago the price of aluminum was \$5 per pound. Now it can be purchased for 50 cents per pound. With the reduction in price has come a gradual employment of the metal in all lines of machine work. The writer has interviewed knitting machine builders who have experimented with aluminum and aluminum bronze castings in parts of knitting machinery that require great strength, and other builders who are contemplating work on this line.

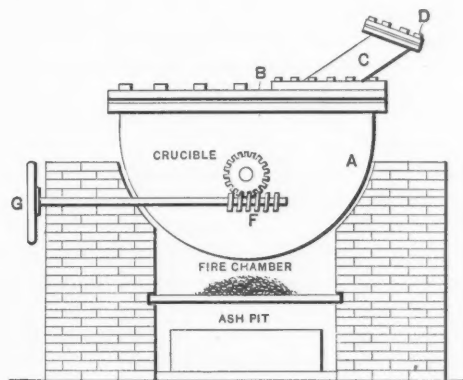


Fig. 1

As the processes of treating and casting aluminum for knitting machinery parts are quite new, the accompanying illustrated description may be useful to those knitting machine builders who wish to try the new metal. Aluminum castings can be bought at the foundries from 45 cents per pound and upward. The prices for aluminum hardened with other alloying metals is a few cents higher per pound. The tensile strength of the hardened metal averages 21,000 pounds per square inch, thus making it well adapted to many small parts of the knitting machine. Tests have demonstrated that the shrinkage of aluminum is about one-quarter of an inch in a twelve inch sample. Although advisable to buy the aluminum crude castings already hardened, the purchaser may increase the hardness by adding zinc, manganese, tin, brass, nickle or other like metals. Aluminized zinc brass makes a most durable metal for knitting machine needle cams.

### PROCESS OF MAKING ALUMINUM CASTINGS FOR KNITTING MACHINERY.

Electrical machinery builders are now using this metal in places where strength is needed. For general purposes, however, a good alloy from which to make cams, bearing sleeves, small castings and the like for knitting machinery may be made as follows:

Aluminum.....	95.
Antimony.....	3.04
Tin.....	0.96
Copper.....	0.94
Tungsten.....	0.06
Total.....	100.00

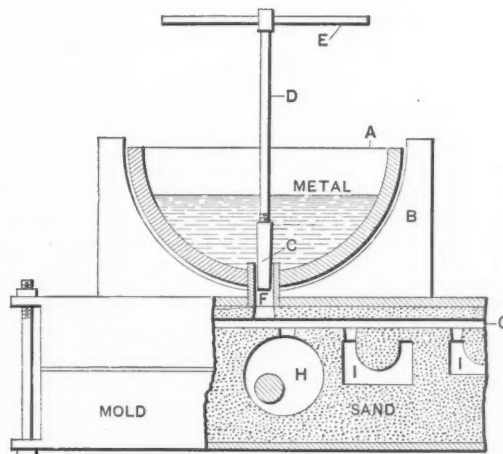


Fig. 2

The tensile strength of parts made from this alloy is 26,000 lbs. It can be cast in sand molds. The melting and casting of aluminum alloys into knitting machinery parts requires an equipment something like that shown in the diagrams, the first

of which is a sectional view of the melting pot, or crucible A, which is set up in a brick affair over a coke fire. Coke must be used, and not coal, as the gases from the latter are liable to impregnate the metal. The crucible can be balanced on a shaft, on which shaft should be keyed a gear. A worm shaft can be arranged to engage with the cogs as shown at F, and by turning this shaft with a handle at G, the crucible can be poured. The lid B is provided with a spout C, the end of which can be closed with the cap D. The latter is, of course, removed when the metal is poured. Next the ladle should be erected as shown in section in Fig. 2 in which the ladle itself is marked A, and the pouring gate channel F. This gate is stopped with the plug C, and the latter raised and depressed by means of the shaft D and handle bar E. By this arrangement the metal is poured from the bottom, and scum is avoided. The arrangement of the mold does not differ enough from ordinary descriptions to call for detailed account.

The gate channel leads into the runner G, and the metal flows through to the molds H, and I. Perhaps in this connection it may be well to refer to the method of casting aluminum shuttles. The mold box is shown in sectional view in Fig. 3 and the gate for pouring is at C, where it branches off into the runner, and several shuttle molds are connected with it. The core of the hollow of the shuttle is indicated A, consisting of a correctly shaped metallic piece, wound with hemp and swept with loam. It is held in place by the stem B. After the castings are taken from the molds, the finishing processes call for the usual chipping, cleaning and smoothing. Some special wheels are used in the work, and these are shown in the next sketches, that in Fig.

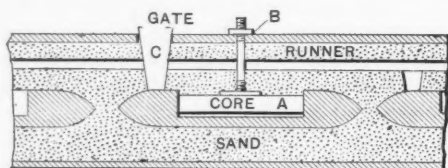


Fig. 3



Fig. 4



Fig. 5



Fig. 6

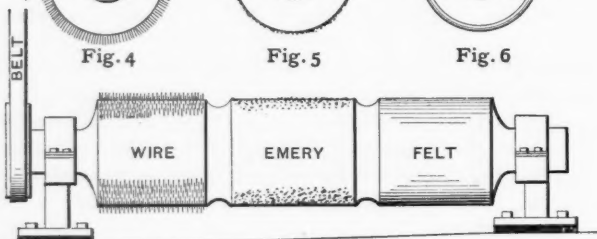


Fig. 7

4 being a side view of a "scratch" wheel made by covering a wood cylinder with medium sized fancy clothing wire. The use of this wheel is to remove fins, etc., from the castings. The next wheel (5) is covered with fine emery, for evening off the surfaces and the last wheel (Fig. 6), is for buffing purposes, being covered with felt or pile fabric.

These wheels may be set up separately in stands of wood and revolved by belts, or all may be arranged on one cylinder with a division between each, as shown in Fig. 7.

If the castings should prove too brittle, anneal them by heating to cherry red and dipping in cold water. Three points are claimed for the aluminum hollow castings. They are lighter, tougher and look better than iron or steel castings. On the other hand, they cost more, but not much more than iron or steel parts. Concerning looks, this is quite an item in these days, and if the aluminum alloy castings are given the so-called nickle polish, a very attractive machine is made. This nickle polish is easily and cheaply applied, consisting in buffing the surfaces of the castings or other aluminum parts, and putting on nickle rouge or crocus composition. The buffing is done in this case with soft muslin or loose cotton wheels. A shiny-nickle-like polish results and sets off the kitting machine to advantage.

\*\*\*

Is not a ball seat for a safety valve the best seat in the world? If not, why not? Ought the angle of the seat—the average angle—to be 45°? If not, what should it be?

## MECHANISM FOR MECHANICS.—6.

PROF. CHAS. H. BENJAMIN.

### INVOLUTE TEETH.

Involute, or curves described by a point in a straight line which is rolling on a circle, are used at present for gear teeth more than cycloids.

It is necessary to prove that curves of this kind will also work smoothly and give a uniform velocity to the follower. In Fig. 20 let C and D be the centers of two gears and C X and D X the radii of their pitch circles such that  $CX = \frac{1}{2} DX$ .

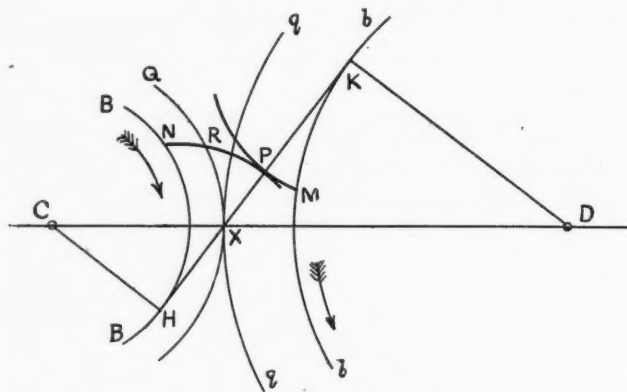


FIG. 20.

Through X draw the indefinite oblique line H K. Tangent to this line describe two circles B B and b b, having radii C H and D K, and suppose that these circles are to be used for generating the involutes for the teeth. These circles are usually called *base circles*.

Imagine the line H K to be a fine cord connecting the two circles. Then as the circle B B turns as shown by the arrow, the cord will wind on B B and off b b, turning the latter circle with a rim speed the same as its own.

The line K H will always pass through X, and as the triangle C H X is similar to D K X,

$$\frac{CH}{DK} = \frac{CX}{DX} = \frac{1}{2}.$$

Hence as D K is twice C H, the circle b b will only turn one-half as fast as the circle B B, and the angular velocities will be the same as if the two pitch circles Q Q and q q were rolling together.

Now let a point P travel along with the line K H. As the line is at the same time winding off b b and on B B, the point P would trace on a surface which is turning with B B an involute P N and on a surface which turns with b b another involute P M.

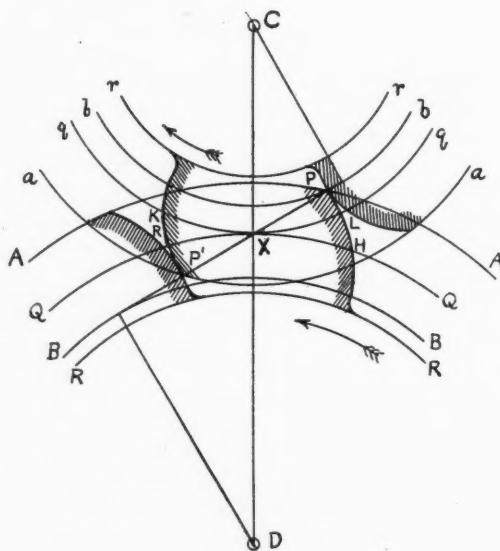


FIG. 21.

The two involutes will always touch wherever P happens to be on the line K H, and this line will be the common perpendicular of the two curves. Now if we use these two curves for outlines of teeth of the two gears, the common perpendicular to the teeth at the point of contact will be K H, always passing through X

and therefore the angular velocity ratio of gear D to gear C will be constant and equal to one-half as desired.

It will be seen that the involutes will lie both inside and outside the pitch circles; for instance, the involute P N has the part P R outside the pitch circle Q Q and the part R N inside. P R is a face and R N a flank, so that in this system the face and flank form one continuous curve, instead of two curves as in the cycloidal system.

In Fig. 21 are shown the pitch, addendum, base and root circles of the two gears having involute teeth. The proportions are the same as in Fig. 15 for cycloidal teeth, the idea being to compare the two forms.

During the approach the flank P L of the driver pushes the face P H of the follower to X, the contact between the teeth always being somewhere on the line P X. During the recess the face K P<sup>1</sup> of the driver pushes the flank R P<sup>1</sup> of the follower from X to P<sup>1</sup>, the point of contact always being on X P<sup>1</sup>.

As in Fig. 15 the arc L X = H X is the arc of approach and the arc X K = X R is the arc of recess. The line P X P<sup>1</sup> is the line of pressure on action between the teeth and always passes through X. In this case it is also the path of contact. The portion of the flank of each tooth which is inside the base circles B B on b b is for clearance, and never touches the teeth of the other wheel. It may be simply a radial line with a fillet for strength, where it joins the root circle.

#### DIFFERENCE BETWEEN INVOLUTE AND CYCLOIDAL TEETH.

By referring to Figs. 15 and 21 the principal differences between the two curves may readily be seen.

1. The involute tooth is of one curve and the cycloidal of two curves. The involute is easier to cut, and on small pinions is broader at the base and stronger.

2. The line of pressure P X P<sup>1</sup>, in Fig. 21, makes always the same angle with the line of centers, that is the obliquity of the pressure is always the same.

In Fig. 15 the lines P X and X P<sup>1</sup> are continually varying in angular position, the angle C X P being the least in the position shown and becoming 90° when the tooth reaches X.

In involute teeth there is always considerable pressure tending to force the wheels out of gear on account of this constant obliquity, and this is a disadvantage when the gears are slow-running and doing heavy work. On light, quick-running gears this may be an advantage since it will tend to prevent chattering.

3. With cycloidal gears it is absolutely necessary that the line

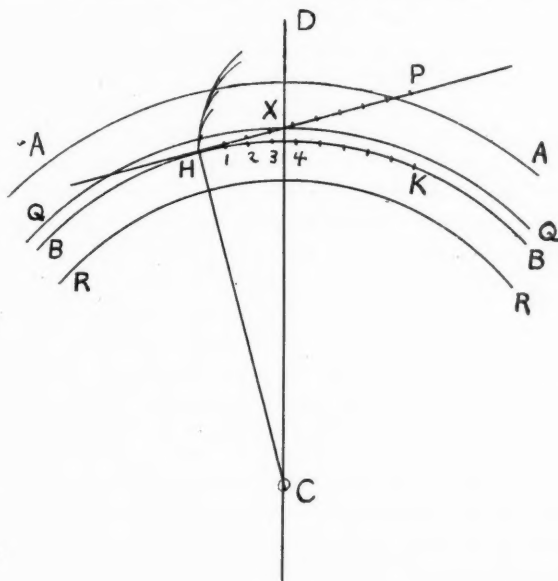


FIG. 22.

of centers should be exactly right, so that the pitch circles may just touch or the teeth will not work correctly.

Involute gears, on the other hand, may be moved to or from each other a certain distance without affecting the smoothness or correctness of the action. By reference to Fig. 20 it may be seen that the only effect of moving the base circles further apart will be to change the angle of the line of action K H. This line will still be tangent to both circles and still pass through X, while the form of the teeth will remain the same as long as the base circles do not change in size. This property of involute

teeth makes them particularly suitable for change gears on lathes and other machines where the distance between centers is likely to be changed.

#### HOW TO DRAW INVOLUTES.

The involute may be considered either as generated by a point in a flexible string unwinding from a circle or by a point in a straight line rolling on a circle, one operation being the same as the other as far as the tracing point is concerned.

In Fig. 22 let C be the center of the pitch circle Q Q of a gear and let A A and R R be the addendum and root circles of the same gear. Draw the line P X H making the angles C X H = 75°. From C drop the perpendicular C H on P X H and with a radius = C H describe the base circle B B just touching P H. Take a portion H P of this line such as to reach beyond the circle A A and take an arc H K of the base circle equal to H P.

Divide H P into a number of equal parts and H K into the same number of equal parts. Take centers at 1, 2, etc., and with radii equal to corresponding parts of H P describe a series of short arcs touching each other as shown. A curve drawn touching the outsides of these arcs will be the required involute. This construction is almost identical with that shown in Fig. 17 for cycloids, but in this case the whole working outline of the tooth is obtained at one operation. The portion of the tooth inside the base circle is usually a radial line with fillet, as before explained. A template of the tooth curve may be made as in the case of cycloidal teeth, and the rest of the teeth drawn in that manner.

#### INVOLUTE RACK.

The teeth of the involute rack are bounded by straight lines which make the same angle with the pitch line as the line of action makes with the centers, usually 75 degrees.

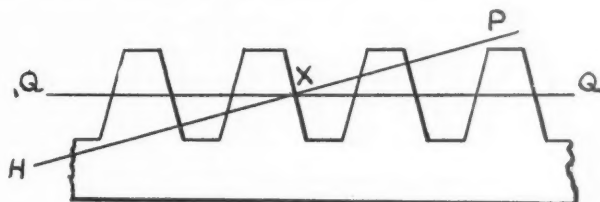


FIG. 23.

As shown in Fig. 23, P X H is the line of action and corresponds to the same line as used in the preceding figures, and Q Q is the pitch line of the rack. Now as the faces of the teeth in any position must be perpendicular to P X H, it follows that they must be straight lines as shown. This simplicity of the rack tooth is a marked advantage of the involute system.

\* \* \*

#### GROWTH OF ENGINEERS' SOCIETIES.

The rather remarkable, not to say phenomenal, growth in this country of the two great societies of stationary engineers—the N. A. S. E. and the A. O. of S. E.—brings up vividly what can be done by common enough men working to an honest end with an honest purpose. Without any other purpose than the education of their members, these associations have advanced the standing of every individual member. I happen to have before me at this writing the preamble of the N. A. S. E. It is a model in its way:

"This Association shall at no time be used for the furtherance of strikes, or for the purpose of interfering in any way between its members and their employers in regard to wages; recognizing the identity of interests between employer and employee, and not countenancing any project or enterprise that will interfere with perfect harmony between them.

Neither shall it be used for political or religious purposes. Its meetings shall be devoted to the business of the Association, and at all times preference shall be given to the education of engineers, and to securing of the enactment of engineers' license laws in order to prevent the destruction of life and property in the generation and transmission of steam as a motive power."

This association has arrived at its present standing by a close adherence to this preamble. I speak particularly of the N. A. S. E. because of being better acquainted with its purposes and for no other reason. There is good reason to believe that these two associations will soon unite, forming one of the greatest educational associations in the country. The work of such an association will not only be a benefit to its members, but to the country at large.

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Entered at the Post-Office in New York City as Second-class Mail Matter.

# MACHINERY,

A practical journal for Machinists and Engineers,  
and for all who are interested in Machinery.

PUBLISHED MONTHLY BY

THE INDUSTRIAL PRESS,

411 AND 413 PEARL STREET, NEW YORK CITY.

ONE DOLLAR A YEAR, POSTAGE PREPAID, TEN CENTS A COPY.

FOREIGN SUBSCRIPTIONS ONE DOLLAR AND FIFTY CENTS A YEAR.

Lester G. French, Editor.

F. F. Hemenway, Consulting Engineer.

Walter Lee Cheney,

S. Ashton Hand,

Associate Editors.

The receipt of a subscription is acknowledged by sending the current issue. Remittances should be made to THE INDUSTRIAL PRESS, and not to the Editors. Money enclosed in letters is at the risk of the sender. Changes of address must reach us by the 15th to take effect on the following month; give old address as well as new. Domestic trade is supplied by the American News Company or its branches.

We solicit communications from practical men on subjects pertaining to machinery, for which the necessary illustrations will be made at our expense. All copy must reach us by the 10th of the month preceding publication.

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AUGUST, 1897.

## CONTENTS:

Notes from the Builders' Iron Foundry..... 359

Position of Draughtsman..... 360

G. Edw. Smith 360

Outlook for American Machinery in Great Britain..... 362

Marine Engine Design (2), Wm. Burlingham 363

Notes from a Clock Factory, Fred H. Colvin 366

Practical Pattern Making (3), I. McKim Chase 367

Making a Propeller Wheel Pattern, Geo. W. Dean 368

Aluminum in Knitting Machinery, Geo. D. Rice 369

Mechanism for Mechanics (6), Prof. Chas. H. Benjamin 370

Mechanical Text Books. Ed..... 372

Poor Work in the Shop. Ed..... 372

Mechanical Draft (1), Walter B. Snow 373

How to Calculate, Design and Construct Electrical Machinery (3), Wm. Baxter, Jr. 374

From Actual Practice..... "Milo" 377

Spur Gear Arithmetic (2), A. B. Babbitt 378

The Hartford Meeting..... 380

Seen in Various Shops..... 383

Valve Gears (9)..... E. T. Adams 384

Graphic System of Computation, "Cismarf" 385

The Split Pulley Question Again..... 386

What Mechanics Think..... 387

How and Why..... 387

## MECHANICAL TEXT-BOOKS.

The best statement that we have seen of what mechanical text-books ought to be is contained in the preface to the well-known treatise, "Theoretical Mechanics," by Professor Weisbach. Coming from so high an authority and in connection with a work, which, at its best, must be almost entirely of a theoretical and mathematical nature, it is significant. He says:

"If we consider how many subjects a technical man must master in order to accomplish anything very important in his profession, we must make it our business as teachers and authors for technical men to facilitate the thorough study of science by simplicity of diction, by removing whatever may be unnecessary, and by employing the best known and most practical methods. For this reason I have entirely avoided the use of the calculus in this work."

By quoting this extract we do not wish to be understood as condemning the use of the higher mathematics. They

have a place and are both useful and necessary. In too many technical books, however, the rules and formulas are arrived at by mathematics too difficult for the average reader to follow, and then left without enough explanation to make them of much value to anyone except students and professors.

Explanations ought to be full and explicit. If advanced mathematics are necessary in deducing results, the results themselves can be easily put into such shape and illustrated with such examples and solutions that they can be used by men who are unable to follow the mathematical reasoning. Some of our best writers do this, and others do not. It does not detract from one's reputation as a scholar or an authority to make his writings clear. It adds to it.

\* \* \*

WHAT are the objections to an effective, equitable license law, all over the country, for stationary engineers? We do not say that it is a good thing. It looks as if it would be, but we are open to information and conviction. It seems that it would lead to an improvement in the class of men employed as stationary engineers, and that would be a good thing to do. We shall be glad to hear from any reader on the subject. Something good ought to grow out of many opinions.

\* \* \*

## POOR WORK IN THE SHOP.

One of the most troublesome questions that has to be decided in the machine shop is what to do with bad castings and spoiled work, or with work, which, while not exactly spoiled, is not right and as it ought to be.

When this question comes up, it seems to be in human nature to scheme on some method by which the casting can be saved, or the work made to go. Especially is this so on large work. It requires more courage to throw out a casting weighing 500 pounds, on which a few hours machine work have been spent, than to condemn a small piece representing several days of labor.

If this course is followed as a regular thing, it is sure to lead to grief sooner or later. Work which is inferior and defective will come to light sometime and in a very embarrassing way for those concerned.

There are cases, of course, where the defect amounts to so little that it would be folly to condemn the piece, and oftentimes the job is so valuable that it would be out of the question to throw it out. Concerning these cases, there is generally small chance for hesitation, and it is evident at once what is the best thing to do. Where there is doubt, however, and there seem to be arguments on both sides, a good rule to follow is to throw the work out. A foreman or superintendent, with backbone enough to stand up to this rule, will be on the right track twice where he is wrong once. Moreover, a quick decision to this effect will save time lost in keeping men and machines waiting, and will increase the respect of all concerned for the management, the methods and the quality of the work turned out.

Another phase of this subject comes up when it is intended to have the work interchangeable and it is found that a piece of work is not made to dimensions. The first impulse is to make the other piece to fit and keep a record of the size, which, if it is filed away, is likely to be overlooked when the first repairs are ordered. Within reasonable limits, the only safe way to do is to make the piece right that is wrong. Two wrongs in machine work do not make a right, and the more firmly this is impressed, the better.

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THE series of articles upon Mechanical Draft, by Mr. Walter B. Snow, the first of which appears in this issue, will be of timely interest to all who have to do with steam generating plants. As far as we know, it is the first adequate treatment of the subject by any journal.

MECHANICAL DRAFT.—I.

WALTER B. SNOW.

With the increasing adoption of mechanical draft as a substitute for chimney draft, its nature and advantages are becoming more widely understood. It is to-day hardly necessary to define the term mechanical draft, but in its common acceptance it may be described as the method under which a fan is employed to produce the draft necessary for combustion in steam boilers. For a full understanding and discussion of the subject it is necessary to treat briefly, at least, of fuels, their composition and the phenomena of their combustion.



Although the term combustion properly applies to all forms of chemical union, it is usually employed only with reference to the process of burning, whereby a substance unites with oxygen, with resulting light and heat. Carbon, the most common constituent of all fuels, concerns most intimately any discussion of the subject of combustion. When burned in the presence of an excess of oxygen, as is the case when an ample supply of air is provided, the gaseous compound known as carbonic acid is formed. This consists by volume of one part of carbon and two parts of oxygen, while by weight its composition is 12 parts of carbon and 32 parts of oxygen.

If there be a lack of oxygen the combustion is incomplete. The carbon unites with only one part of oxygen, forming carbonic oxide, which consists by weight of 12 parts of carbon and 16 parts of oxygen. Associated with the carbon of practically all fuels is a greater or less amount of hydrogen, which upon burning unites with the oxygen in the proportion of two parts by volume of hydrogen and one of oxygen, to form water, and by weight of two parts of hydrogen and 16 parts of oxygen. From a knowledge of these resulting combinations of carbon, and hydrogen with oxygen, it is possible to determine the amount of air chemically necessary for the complete combustion of one pound of coal if its composition be known.

For simplicity of expression, the elementary substances are indicated by symbols, preferably their initial letters, as for instance, carbon by C, oxygen by O, nitrogen by N and hydrogen by H. The volumetric combinations of these elements are likewise represented by combinations of the symbols. Thus the compound, carbonic oxide, consisting by volume of one part of carbon and one of oxygen is designated by CO; carbonic acid, having two parts of oxygen to one part of carbon, is designated CO<sub>2</sub>, the suffix "2" serving to indicate the relative number of atoms of oxygen. In a similar manner the symbol of water becomes H<sub>2</sub>O.

For the combustion of one pound of carbon there is required 2½ pounds of oxygen; for, as already shown, carbonic acid, the

product of complete combustion, consists of  $\frac{12}{32} = \frac{12}{44}$  of carbon and  $\frac{32}{44} = \frac{32}{44}$  of oxygen; hence the proportion of carbon

is to that of oxygen as 12 : 32 = 1 : 2½. As air consists by weight of 0.236 parts oxygen and 0.764 parts nitrogen, in a mechanical mixture, it will require 2½ ÷ 0.236 = 11.3 pounds of air to supply the oxygen required for the combustion of one pound of carbon.

In a similar manner it can be shown that one pound of hydrogen requires 8 pounds of oxygen. Making due allowance for inherent oxygen in the fuel, the total ideal requirements in the way of air may be calculated for any given composition of fuel, a simple approximate formula taking this form:

$$\text{Weight of air} = 12 C + 36 \left[ H - \frac{O}{8} \right]$$

in which the weight of carbon is represented by C, that of hydrogen by H, and that of oxygen inherent in the fuel by O. A fair average coal will show the weight of air required to be about 12 pounds.

Such calculations are based upon the assumption that each individual atom of oxygen comes in contact and unites with its proportion of hydrogen or carbon in the fuel. When it is considered, that this oxygen is intimately mixed with about four times its volume of nitrogen, whereby it is to a certain extent separated from the fuel, and further, that the variety in the arrangement of the fuel and the passages through it affect any attempt at equal distribution of the air, it must be evident that the above assumption cannot ordinarily be maintained in practice. It therefore becomes necessary in practice to furnish sufficient air in excess of the calculated amount to insure complete combustion in all parts of the furnace.

Evidently the amount of air supplied for dilution must vary greatly in different cases. Thus in the series of comparative tests reported by Messrs. Donkin and Kennedy, the dry air supplied per pound of coal ranged from 16.1 pounds to 40.7 pounds, and the corresponding ratio of air used, to air theoretically required, varied from 1.56 to 4.28; that is, from 56 per cent. to 328 per cent. in excess.

The recent tests of Mr. J. M. Whitham upon automatic mechanical stokers have shown most remarkable results. With a combustion of 12 pounds of buckwheat coal per square foot of grate per hour, the air supply, with a Wilkinson stoker and forced draft, was found to be 85.6 per cent. in excess of that chemically required; while with a rate of 45.4 pounds, almost perfect evaporative efficiency was secured when there was an actual deficiency of 11.2 per cent. in the air supply below the chemical requirements.

Accepting 12 pounds of air as, in round figures, the amount theoretically necessary, the weight and volume, with 50 per cent. and 100 per cent. dilution—the air being at 62°—are as shown in Table No. 1.

TABLE NO. 1.—AMOUNT OF AIR REQUIRED FOR COMBUSTION.

	Without Dilution.	With 50 per ct. Dilution.	With 100 per ct. Dilution.
Weight of air.....	12 pounds.	18 pounds.	24 pounds.
Volume of air, exact.....	157.7 cubic feet.	236.5 cubic feet.	315.4 cubic feet.
Volume of air in round numbers.	150 "	235 "	300 "

An insufficient supply of air causes imperfect combustion of the fuel, which in bituminous coal is indicated by the production of smoke, and in coke and anthracite coal by the discharge of carbonic oxide from the chimney. An excess of air causes waste of heat corresponding to the weight of air in excess of that which is necessary, and to the elevation of temperature at which it is discharged from the chimney above that of the external air. Evidently great losses may result from the admission of too great an excess of air, whence any means by which it may be more advantageously used will tend to economy.

The true economic working of the fuel, as regards the completeness of its combustion, is most clearly evidenced by an analysis of the flue gases, whereby are determined the relative proportions of carbonic acid, carbonic oxide and oxygen in the gases leaving the boiler. From such an analysis the amount of nitrogen and of air supplied may be determined. By analyzing the gases taken from different points in their progress, as for instance, just back of the bridge wall and again where they enter the stack, the amount of leakage or infiltration may be accurately ascertained.

By the *heat of combustion* is measured the theoretical heating value of any given substance. As determined by the most recent and refined calorimetric tests, and expressed in British thermal units, the heat of combustion of the important substances concerned in combustion is as given in Table No. 2.

TABLE NO. 2.—HEAT OF COMBUSTION OF SUBSTANCES.

Substance.	Heat of Combustion in British Thermal Units.
Carbon burned to CO <sub>2</sub> .....	14 650
Carbon burned to CO.....	4 400
Hydrogen.....	62 100
Marsh Gas.....	23 513
Olefiant Gas.....	21 343
Carbonic oxide burned to CO <sub>2</sub> .....	4 393

The great loss of heat due to the incomplete combustion of carbons, is clearly presented in the differences between the total heat of perfect combustion of carbon to  $\text{CO}_2$  (viz., 14 850 B. T. U.), and that of carbon to  $\text{CO}$  (viz., 4 400 B. T. U.), the latter being the product of incomplete combustion as already stated.

The total heat of combustion of various American and foreign coals, as given in Table No. 3, shows that as regards the better grades there is comparatively little difference.

TABLE NO. 3.—HEAT OF COMBUSTION OF FUELS.

Kind of Fuel.	ANALYSIS.					Caloric value Calculated, B. T. U.
	Carbon.	Hydrogen.	Oxygen and Nitrogen.	Hydroscopic Water.	Ash.	
Anthracite from Pennsylvania.	86.456	1.995	2.199	3.450	5.900	15 210
Semi-anth. from Commentry.	84.928	2.892	5.005	1.775	5.400	15 048
Semi-bituminous from Aniche.	85.937	4.198	5.240	0.625	4.000	15 638
Bituminous from Auzin.	83.754	4.385	5.761	1.100	5.000	15 640
Wigan cannel coal.	78.382	5.060	5.058	0.600	10.900	16 220
Lignite from Styria.	65.455	4.782	24.303	0.710	4.750	11 898
Coke, Pennsylvania anthracite.	91.036	0.685	2.146	0.233	5.900	14 540

The calorimetric and calculated values of the heat of combustion usually agree quite closely, but in boiler practice, owing to the opportunities for loss of heat through radiation, heat carried off by flue gases, incomplete combustion, etc., the maximum efficiency attainable falls short of that indicated by the total heat of combustion. Even with the best possible boilers and warm-blast or feed-water heating, 90 per cent. efficiency appears to be about the limit, while under ordinary conditions with good coal, the average may be assumed at from 60 to 70 per cent., and with poor coals at from 50 to 60 per cent. It is, therefore, customary for rough figuring and fairly good conditions to consider the available heat per pound of fuel as ordinarily from 10 000 to 12 000 B. T. U.

From the total and specific heats of combustibles may be calculated the temperature which would result from their combustion if all possible losses were prevented. In ordinary practice these losses must occur, and the efficiency of the fuels be reduced thereby. It is therefore impossible in practice to obtain the full ideal temperature.

One of the most important factors is the determination of the temperature of combustion which will result under given conditions, of the volume of air which is supplied to the fuel. Even the inert nitrogen which accompanies the oxygen actually required for complete combustion has its influence, and a most important one, in reducing the ideal temperature of combustion with only the requisite volume of oxygen, for it must all be heated to the maximum attainable temperature. Evidently, therefore, any excess of air must have a most marked effect upon the resulting temperature. This is clearly evidenced in Table No. 4, as calculated for a standard semi-bituminous coal.

TABLE NO. 4.—IDEAL TEMPERATURE OF COMBUSTION WITH DIFFERENT DEGREES OF DILUTION.

Percentage of Dilution.	Ideal Temperature.	Loss of Temperature due to Dilution.
0	4718°	
50	3353°	1365°
100	2600°	2118°
150	2124°	2594°

This table plainly shows the economy, other things being equal, of supplying as little air as will successfully produce complete combustion of the fuel.

The ultimate efficiency of a fuel should be expressed by the total amount of heat it is capable of generating. The proportion of that heat which is utilized depends upon the efficiency of the boiler or other heat-abstracting device. Commercially, however, the heat value of fuels is generally measured relatively to each other, and is expressed in the number of pounds of water evaporated per pound of fuel. In practice the physical character of the fuel, the form and construction of the boiler and furnace, the amount of air supplied, and the means adopted to provide it, have an important influence upon the attainable results.

A mere comparison of the amount of water evaporated per pound of different kinds of fuel, is not sufficiently exact to warrant confidence, for there may be great variations in the conditions of steam pressure and temperature of feed-water under which the tests were made. It is therefore customary to reduce all results to a common basis, namely, to a condition of feed-

water at 212° and steam of atmospheric pressure, that is of a temperature of 212°. Hence the common expression, "from and at 212°." Under these conditions no heat is expended in heating the water, so that the amount of heat required to evaporate one pound of water is exactly equal to the latent heat of steam at atmospheric pressure; that is, 965.7 B. T. U. The evaporation per pound of coal or combustible, as reduced to this basis, is known as the *unit of evaporation*.

For a clear comprehension of the possibilities of increased economy, resulting from the application of mechanical draft, consideration should be given to the losses both avoidable and unavoidable, which do or may take place in connection with the combustion of fuel. The unavoidable losses are:

*First.* Those resulting from converting into steam the water in the coal and in the air used in burning it, as well as the burning of the hydrogen and the heating of all of the steam to the temperature of the escaping gases.

*Second.* Those incident to heating to stock temperature the carbonic acid, the nitrogen and the excess of air. Also those from hot ashes and carbon therein, as well as from radiation through boiler covering or setting. These latter can be reduced, but not eliminated.

The more or less avoidable losses are:

*First.* Those due to incomplete combustion.

*Second.* Losses from excess of air.

*Third.* The result of too high temperature of the escaping gases.

*Fourth.* The waste through hot ashes and carbon therein.

*Fifth.* Loss by radiation.

The influences which these various factors exert upon the ultimate efficiency of a boiler, the possibilities of their reduction and their relation to the introduction of mechanical draft, will be considered in the next number.

\* \* \*

## HOW TO CALCULATE, DESIGN AND CONSTRUCT ELECTRICAL MACHINERY.—3.

WM. BAXTER, JR.

In the preceding article it was shown that when a wire moves across a magnetic field, an electro-motive force is developed in it, and that if there is no current from an external source in the wire, or a current in the same direction as that induced by the motion, the wire acts as a generator; but if there is an external current, and this flows in the opposite direction to the induction, the wire acts as a motor. The manner in which the lines of force were drawn to represent the action, in both cases, is not in accordance with the plan generally followed, but it was used because it illustrates the action in a more comprehensive manner. The general explanation given in text books is that the cutting of the wire across the lines of force sets up an electro-motive force, but the diagrams furnished to illustrate this action do not impress the eye with the fact that in one case there is a resistance to motion, while in the other there is an actual pull to produce motion. It makes little difference, however, how the picture is drawn, so long as we remember that the effect is caused by the cutting of the wire across the magnetic field. The lines of force, as has been said, are purely a picture of the imagination, to assist us in understanding the action; but the term is also used as the name of the unit of magnetic force, and as such it is far from being imaginary. In this sense it is very real, as it represents a definite amount of force.

The unit of electro-motive force, or electrical pressure, is the volt, the unit of electric current is the ampere, the unit of resistance is the ohm, and unit of energy, or power, is the watt.

A volt is that amount of electro motive force that is necessary to force a current of one ampere across a resistance of one ohm; an ohm is that amount of resistance to the passage of a current that is sufficient to keep the strength down to one ampere, when the pressure of the current is one volt. A watt is the amount of energy represented by a current of one ampere under an electro-motive force, or pressure, of one volt.

When a wire cuts across a magnetic field, an electro-motive force is developed in it, and the value of this, measured in volts, is independent of the strength of the current; hence the calculations we make in regard to the effect of rotating one, or any number of wires in a magnetic field, have for their object the determination of the electro-motive force, in volts, that may be

developed by the action. The strength of current that may be obtained with this electro-motive force will be dependent upon the resistance which opposes its flow, and by varying this resistance we can vary the current.

In order to be able to determine what voltage will be developed in a number of wires rotating in a magnetic field, we must know the relation that exists between the strength of the field and the electro-motive force developed, by the movement of a wire through it. This relation is known, and it has been demonstrated by theoretical reasoning and by actual experiment, that it is unvarying; that is, that the electro-motive force developed by the cutting of one line of force in a given time by one wire, is under any and all conditions the same.

The line of force when used as a measure of magnetic force represents a very small quantity, and on this account many attempts have been made to bring into use a larger unit of measurement; but although many have been proposed, the line of force still remains the universal standard. The force that it represents is so small that it is necessary to cut one hundred million lines per second to develop an electro-motive force of one volt. *If you have a magnetic field in which there are one hundred million lines of force and one wire is moved across this field in one second, the electro-motive force (which is represented by the letters E. M. F.) will be one volt, and it will be maintained during the one second in which the wire is moving through the field. If the wire is kept moving continuously through the field at the same rate, i. e., one passage per second, a continuous E. M. F. of one volt will be developed. If the wire moves across the field in half a second, the E. M. F. will be two volts; if it moves across in five seconds, the E. M. F. will be one-fifth of a volt. If two wires are moved instead of one, the voltage will be doubled and if the number of wires is ten it will be ten times as great.* Studying these statements it will be seen that the E. M. F. is in direct proportion to the number of lines of force cut per second, and that to increase the effect we must increase the number of wires, or the number of times each wire crosses the field, or the number of lines in the field or make an increase in all three. In actual practice the course pursued is to increase the number of wires and the number of times they pass through the field. How this is done can be explained in connection with Fig. 22, which is a simple representation of a two-pole machine. In this cut, the ring R, which is called the armature, carries the wires that cut through the field established by the poles N S, which, with the parts M M' F,

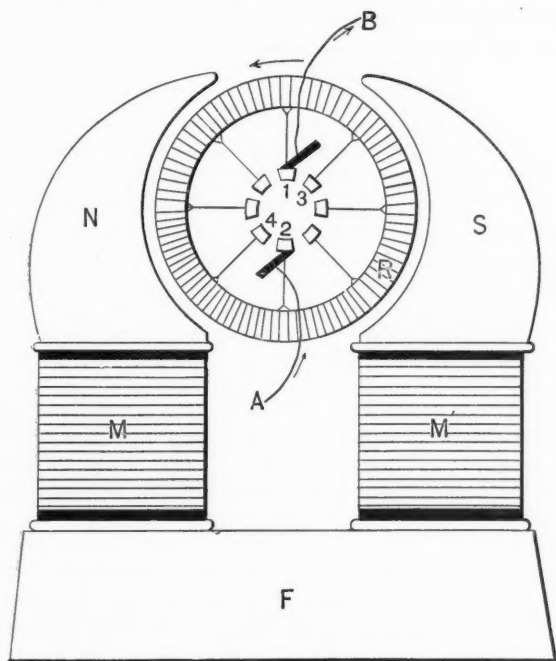


Fig. 22

form a magnet of the horse-shoe type. There are two different kinds of armatures, the drum and the ring; that shown in the figure is of the ring variety. This ring is made of iron, and the wire in which the E. M. F. is developed is wound around it, in a manner more easily understood from Fig. 23. As the ring is of iron, the lines of force coming from the poles N S will not pass into the air space inside of it, but will flow through the ring from

one side of the diameter to the other. On this account the wires on the inside of the ring are not acted upon by the lines of force, and the E. M. F. is induced wholly by the cutting of the outside wires through the field. If it were not for this fact, the machine would not develop an E. M. F., because the induction in the inside and outside wires would be in the same direction with reference to a fixed point, and consequently in opposite directions so far as the wire is concerned; therefore the two actions would offset each other. This can be better understood from Fig. 23. If the lines of force passed through to the inside of the ring, the direction in which the E. M. F. would be developed in the inside wires would be down, if it were down on the outside, but to be in the same direction through the wire it would have to be up on one side and down on the other.

On ring armatures the wire is wound in a continuous coil, as shown in Fig. 23; therefore the E. M. F. induced in the turns covering the two halves opposite the two poles, will be in opposite directions; for, as has been shown, if one side of a loop moves

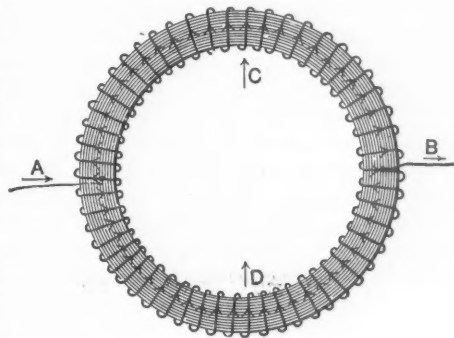


FIG. 23.

through the field in one direction and the other side in the other direction, the inductions in both sides will be in opposite directions. This is indicated in Fig. 23 by the arrow-heads, and by tracing them out it will be found that at the point where the wires A B are attached, the opposing currents meet, and would neutralize each other if it were not for the fact that through these wires the two opposing currents can find an outlet through which they can flow together in the same direction. If the wires A B were connected permanently to the wire wound upon the ring, only a quarter of a turn could be made before the position would be such that the inductions would counteract each other, for, as can be seen, if the wires were connected at the points C D, half the number of turns in each side would be acting against the other half. To get around this difficulty the commutator is used, and the wire wound upon the ring is connected with it in the manner clearly shown in Fig. 22. The wires A B are connected to brushes that are stationary, and set so as to rub upon the surface of the commutator. The wire upon the ring or armature is divided into as many sections as there are divisions in the commutator, but the ends of the sections remain connected so as to form a continuous wire, and from each one of these a wire connection is run down to a commutator segment directly under it. These commutator sections are insulated from each other. As the armature is rotated, say in the direction indicated by the arrow, section 1 will slide from under brush B and section 3 will take its place, and the same thing will happen with respect to sections 2 and 4. So long as the brushes rest upon sections 1 and 2 the current will flow into the armature winding through section 2 and out by section 1, but when the brushes change over to sections 3 and 4 the points at which the current enters and leaves the armature wire will be changed. Thus it will be seen that as fast as the armature rotates, the points at which the current taps the wire is changed, therefore the number of turns of wire in each half of the armature in which the induction is, in the reverse direction cannot be greater than the number of turns in one section, that is, between two consecutive commutator bars, and if the number of bars in the commutator is sufficiently large, this number of wires may be very small.

As was stated in the last article, if we had two coils set at right angles to each other, and each one connected to a separate commutator, a uniform current could be obtained by properly proportioning the machine, but it would be objectionable to use two commutators, and furthermore would be difficult to obtain the exact relation of the various parts, and it would involve complicated calculations. If the poles N S are made so as to cover a large portion of the armature, the current developed in each coil will grow fast and die out slowly, somewhat as shown by the curve A in Fig. 24. If the poles are made so as to cover a small

portion of the circle, the growth and decadence of the current will be more gradual, as shown by curve A in Fig. 25. By obtaining a proper length of pole surface around the circle, the rise and fall of the current can be made to conform to a curve something like that shown at A in Fig. 26. If two currents that pulsate as shown in Fig. 24 were combined, the resultant current would not be uniform, but would be as shown by curve C of Fig. 24, for as one current would be at its strongest point when the other was zero, the resultant current at the line 1 would be only that represented by curve B, and at line L that represented by curve A; but at line D it would be the sum of lines E F and H G, which is K G. Thus the combination of these two currents would result in giving a current that would pulsate twice as fast as each of the currents of which it is composed, but with a slightly smaller fluctuation. If the pole pieces are made very narrow, so as to cause the currents to grow and die out slowly, the current resulting from a combination of the two would be as shown by curve C, in Fig. 25. If the width of the poles were such as to make the growth and decadence of the currents correspond to the curves A B, Fig. 26, the combination of the two would result in a uniform current; but, as stated above, this result can only be obtained by very accurate calculations. From this it will be seen that it would be difficult to obtain a uniform current by the use of two coils with separate commutators. There is another objection to the use of two or any small number

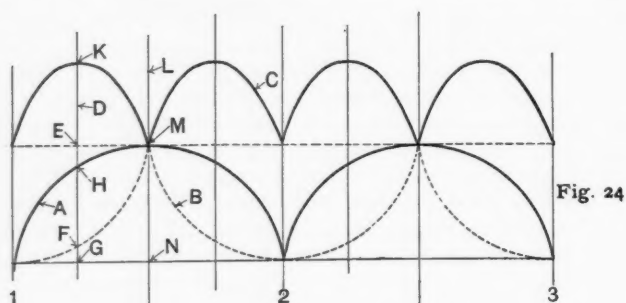


Fig. 24

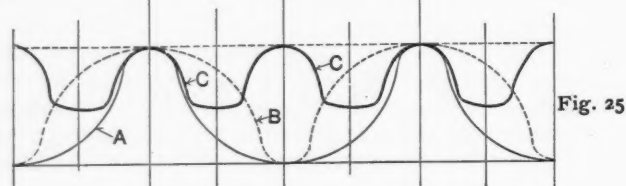


Fig. 25

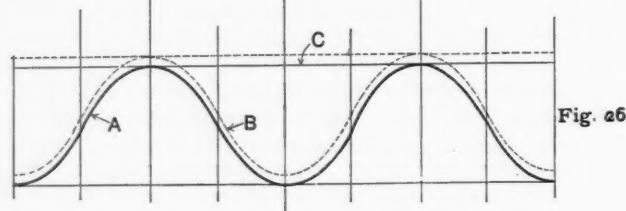


Fig. 26

of coils, and that is that the sparking at the brushes is very much increased thereby. This can be shown by means of Figs 27 and 28, which represent two armature coils of different number of turns. If there are a few coils each one must have a correspondingly large number of turns; the greater the number of coils the fewer the turns in each, for a given number of turns on the whole armature. Now the construction of the commutator must be such that the brush will lap onto one segment or bar before it leaves the other; therefore while it rests upon the two, the coils or sections whose ends connect with these will be short circuited, and as a result a local current will run through the coil, and when the forward segment passes from under the brush, this local current will be broken, and a spark will be produced. If the coil has a few turns, as shown in Fig. 28, the pressure of this local current will be small, being in proportion to the number of turns; but if the coil has many turns, as shown in Fig. 27, the pressure of the current will be much greater, and therefore the spark will be larger, since the length of the spark is proportional to the pressure of the current.

For the reasons given in the foregoing, the wire on the armature is always divided into a large number of coils or sections, except for special cases where, for one reason or another, a departure from this course is necessary.

With all armatures using commutators with a large number of

sections, the connections between the armature and the commutator segment are such that the current must pass through the armature in two paths, as shown in Fig. 23. A study of this figure, as well as Fig. 22, will show that such a result must necessarily follow, because it is not possible to change the points at which the current enters and leaves the wire except by tapping the wire at two points on opposite sides of the diameter. As the



FIG. 28.

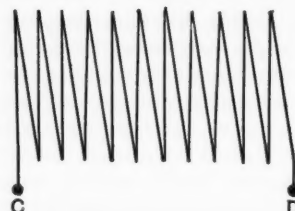


FIG. 27.

current flows in two paths, the E. M. F. induced by the rotation of the armature is that due to the cutting of one-half of the wires through the field, because the two halves of the armature work side by side, or in parallel, as it is termed in electrical phraseology. If the two halves could be connected end to end, then the E. M. F. of the two sides could be added, but not otherwise. From this it will be seen that in making calculations we count one-half the number of wires on the outside surface of the armature, but each wire cuts through the field once in each half revolution; therefore twice as many times in a second as the armature revolves. As multiplying one-half the number of wires by twice the number of revolutions, is the same as multiplying the whole number of wires by the whole number of revolutions, the rule for calculating the E. M. F. developed by an armature rotating in a magnetic field is: *To multiply the total number of wires on the outside surface of the armature by the number of revolutions per second, and by the number of lines of force in the field, and divide this product by 100,000,000, or in other words, point off eight decimal figures.*

This rule, it will be seen, is very simple, and the only precaution to be observed in using it is to point off the proper number of places in each operation; if this is not done, the results obtained may be ten times too great or too small, but if proper care is exercised such mistakes need not be made, and if they are, they can be detected at once after a small amount of experience has been acquired, as then you can judge close enough from the size of the machine to tell whether the figures obtained by calculation are out of all reason or not. If they appear preposterous, go over the calculations again, and the place where the mistake was made will surely be found.

The quantities to be multiplied are only three, and if you are designing a machine you can assume all of them, and if they prove to be too small, you can increase any one or all until the proper size is obtained. As an illustration, suppose you start by assuming that the number of wires on the outside surface of the armature is two hundred, and that the revolutions per second are twenty, equal to 1200 per minute, and that the lines of force in the field are 4 000 000, then

$$\frac{200 \times 20 \times 4\,000\,000}{100\,000\,000} = 160 \text{ volts.}$$

Now if you want the machine to develop an E. M. F. of say 200 volts, it will be necessary to increase all the quantities above the line a small amount or one or two of them to a somewhat greater extent. Thus you could increase the wires to 250 by enlarging the diameter of the armature, or by winding the wire in more layers. You could also increase the revolutions to 25, or the lines of force to 5 000 000. If instead of 160 volts you desire the machine to develop 100 volts, it will be necessary to cut down the figures, and in deciding which to cut, you must use your judgment; and this is the point where the skillful designer shows his superiority.

This rule does not furnish all the information to enable one to design a machine, for after determining the E. M. F. it is necessary to know how much current will be generated in the case of a generator, or what current will be required in the case of a motor. It is also necessary to know how to obtain the required lines of force in the field. All these points will be explained in succeeding articles, and then it will be found that the rules by which these parts of the calculation are worked out are just as simple as the one just given.

## FROM ACTUAL PRACTICE.

"MILLO."

## CAMS—MACHINE SHOP ARITHMETIC—FORMULAS.

In doing a certain job not long ago it became necessary to make a cam that would work in a slot and have no lost motion at any part of the revolution of the cam which moved the slide D, Fig. 3, back and forth between the guides E E. An eccentric changed the motion too slowly, so a cam like Fig. 1 was tried, which is the same shape as that mentioned in the January issue by F. E. Rogers, which reminded me of the use I had previously found for it in actual practice.

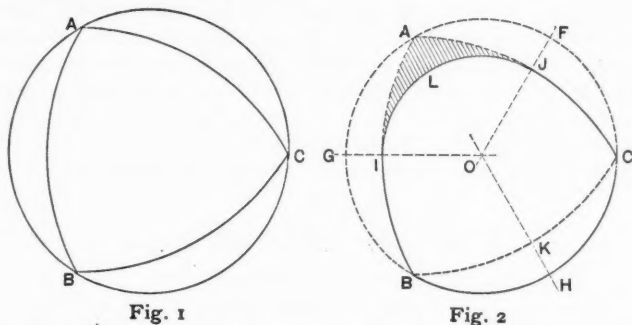


Fig. 1

Fig. 2

With a cam like Fig. 1 it was difficult to stop it when the slide was at the extreme end of the stroke, so the shape shown in Fig. 2 was decided upon, which would start the slide sufficiently quick, wear longer and enable the slide to be stopped at the desired point easily, as it is at rest during  $\frac{1}{8}$  of a revolution of the cam at each end of the stroke.

The shape shown in Fig. 2 is obtained by dividing the circle into six equal parts A G B H C F. With C as a center the arc A I B is drawn, then with B as a center, draw arc A J C. Now, instead of drawing the arc B K C, which would make it exactly like Fig. 1, arc B H C of the original circle is used with the corresponding arc I L J drawn from the same center, O. Arcs A I and A J are not a part of the cam outline, but are drawn to show the construction.

Still another modification of the shape mentioned by Mr. Rogers is shown in Fig. 4 which is sometimes called a square cam as it moves the yoke in four straight lines as indicated by the arrows, the space in which the cam works being perfectly square and the cam is always in contact with each of the four sides of it. While the throw of Figs. 1 and 2 are the same and absolutely determined by the size of the cam the throw of Fig. 4 can be made of any desired amount up to a certain limit.

In drawing Fig. 4 the arcs P N R and S M T, each equal to  $90^\circ$  are drawn from the same center, O; the throw of the cam being determined by the difference between O M and O N just the same as an eccentric.

Then U V and W X are drawn tangent to these arcs and at right angles to M N. U. W. and V. X. are then drawn, forming a perfect square.

Now on each of the diagonals P O T and S O R there is a point O<sup>1</sup> from each of which two arcs P Q and T Z, and R Z and S Q are drawn completing the outline of the cam. These points are determined by the sides of the square O W and V X, and the points P and R.

Points S and T may be used instead of P and R if desired.

Here we have three distinct shapes none of which are round, yet they will all measure the same in whatever direction they are calipered and might easily deceive a fellow who depends entirely upon his micrometer to tell when a piece of work is round.

There are a few problems involved in making the cam shown in Fig. 2, and the mill used, which are of the kind that ought to interest mechanics, even if some of them are mathematical, so I will try to make them plain enough for the apprentices even at the risk of being too elementary.

Referring to Fig. 2. In the first place the diameter of the circle was  $\frac{5}{8}$  of an inch and the arcs A I B and A J C were made

with a hollow mill. Since both arcs are of the same radius only one mill is required and as the arc A I B is drawn from C as a center, the inside cutting points of the mill must describe a circle just large enough so that its center will be exactly at C when the finish cut is made at A I B.

Now the question is, how large must that circle described by the mill be and how can it be found? This is where a little knowledge of arithmetic and geometry comes in handy for the man with the greasy cap and it don't take much time to learn them either.

Now bear in mind that the distance in a straight line from C to A is what we are after as that is the radius of the circle described by the inside cutting points of the mill.

To avoid the lines on the drawing we will work it out in Fig. 5. Connecting A B and C by straight lines we have a triangle with all of its sides equal, an equilateral triangle it is called. Dividing A B at a<sup>1</sup> into two equal parts we know that A a<sup>1</sup>, and B a<sup>1</sup>, are each equal to  $\frac{1}{2}$  of the required distance C A. By geometry we find that a line through one of the points of an equilateral triangle as at C, and cutting the opposite side into two equal parts as at a<sup>1</sup> is at right angles to the opposite side A B, and passes through the center O<sup>1</sup> of a circle drawn through the three points of the triangle. Geometry also proves that a<sup>1</sup>O<sup>1</sup> is exactly equal to  $\frac{1}{2}$  of C O<sup>1</sup> and C O<sup>1</sup> we know to be  $\frac{5}{16}$  of an inch, which makes O<sup>1</sup>a<sup>1</sup> exactly  $\frac{5}{32}$  of an inch.

$\frac{5}{16}'' + \frac{5}{32}'' = \frac{15}{32}''$  which is the distance from C to a<sup>1</sup>.

Of course A O<sup>1</sup> is exactly the same as C O<sup>1</sup>, being drawn from the center to the circumference of the same circle, equal to  $\frac{5}{16}$  of an inch in this case. Now we have got the right angled triangle A O<sup>1</sup>a<sup>1</sup> with two well known elements, a<sup>1</sup>O<sup>1</sup>= $\frac{5}{32}$  and A O<sup>1</sup>= $\frac{5}{16}$  of an inch; and geometry tells us that in any right angled triangle the base a<sup>1</sup>O<sup>1</sup> (shorter leg) squared (multiplied by itself) and subtracted from the hypotenuse A O<sup>1</sup> (side opposite right angle) squared, equals the square of the perpendicular A a<sup>1</sup> (short leg). In other words the result is the product of A a<sup>1</sup> multiplied by itself, so by extracting the square root from this result we get .2706+, which is the distance from A to a<sup>1</sup>; the base of the right angled triangle A C a<sup>1</sup> of which we now have two known elements; the base A a<sup>1</sup>=.2706+, and the perpendicular C a<sup>1</sup> which we have already found to be  $\frac{15}{32}$  or .46875 of an inch.

This distance A a<sup>1</sup>=.2706 of an inch is as we have noticed, just one-half the distance C A that we are after, and we can multiply by 2 and get our results at once; but for the sake of a little practice on figuring triangles, we will not do that.

Now we have got another problem similar to the last, but in this case the perpendicular and base of a right-angled triangle

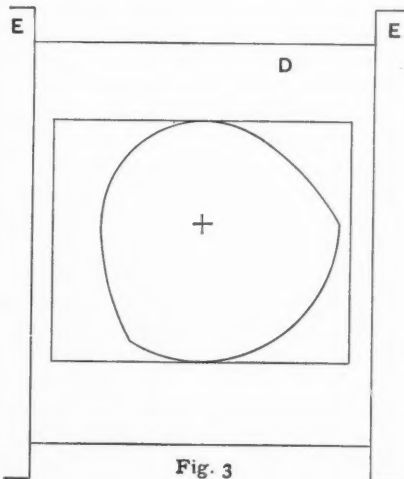


Fig. 3

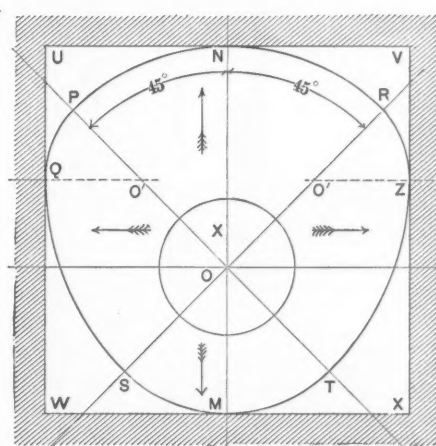


Fig. 4

are the known elements instead of the base and hypotenuse, and geometry says that the square of the base (A a<sup>1</sup>) added to the square of the perpendicular (C a<sup>1</sup>) equals the square of the hypotenuse. By extracting the square root from this square of the hypotenuse we get in this case .5412+ of an inch as the distance from C to A that we are after.

We have now found the radius for the inside of our mill and by doubling it we get 1.0824 inches, which is the diameter of the circle to be made by its inside cutting points.

But a mill never runs exactly true. The milling machine spindle may "run out" or the mill will change in hardening, so we

will make the mill with four points like Fig. 6, having the hollow a little larger than what we need, and a collar with adjusting screws to fit on over it, and close the points in until they describe a circle just the size we want. This plan was suggested by Nay & Co.'s foreman at Hardscrabble, and proved to be a good one.

Now make the piece for the cam, Fig. 7, turning it to size before milling, and set it up in the milling machine so that the center  $O^1$ , Fig. 5, of the work is in line with the center of the milling machine spindle C.

By subtracting A C, which we have found to be .5412 + of an inch, from the diameter of our work, we find that we want to remove .084 of an inch at  $b^1$ , Fig. 5, which is also the throw of the cam.

Holding the work in the spiral head, it is lowered until it will go inside the mill, then raised until it just touches the cutter when it is in motion. The cutter having been previously adjusted to cut to size, we can tell when the proper amount has been removed at  $b^1$  by the index on the machine or by measuring the work, which we have found by the figures wants to be .5412 of an inch from C to I when finished. Indexing the work one-third of a turn, B K C is made in the same way, leaving the shaded portion in Fig. 2 to be removed by some other method. It can be done with the same cutter, but in my case the spiral head didn't run true enough, so I did it with a file, finishing it with a tool in the lathe, having a bolt in the face plate for the dog to strike against to prevent it turning too far, and turning the work backward and forward on the centers by hand. It made a very good job of it.

I would like to start in here and, taking them one at a time in their order, go through all of the problems in geometry from the

very beginning until they lead up to the solutions of the right-angled triangle that we have used, and prove each step, but space forbids.

By looking in some of the hand books for mechanics you will probably find these same two problems that we have worked

out, expressed in the shape of formulas, something like this:  $C^2 - A^2 = B^2$ ; this is the first problem of the right triangle that we solved, and  $B^2$  means the perpendicular squared (multiplied by itself);  $C^2$  means the hypotenuse squared, and  $A^2$  means the base squared. In the second problem the values of the perpendicular and the base were the known quantities, and we wanted to find the

square of the hypotenuse. This would be expressed in a formula thus:  $B^2 + A^2 = C^2$ , the letters meaning the same and the numerical values of any problem we might have can be substituted for the letters and proceed just as we did in the problems already solved. Now suppose the hypotenuse and perpendicular of a right triangle are given, it don't make any difference what their dimensions are, and we want to find the

base. The formula for this would be  $C^2 - B^2 = A^2$ . That is, the square of the perpendicular subtracted from the square of the hypotenuse equals the square of the base, and by extracting the square root from this we get the length of the base.

Here we have got three formulas by which the length of any side of a right triangle can be found if the length of the other two sides are given, and it would be well to remember them.

It makes no difference what characters you use in the formulas to represent the different sides of the triangle, so long as you know what side each one represents. Different books may have different letters to represent the same relative side, but as long as the writer explains what side he means that is all that is necessary. You can use the picture of a corner-stone to represent the base, a plumb-line for the perpendicular, and a side hill for the hypotenuse, if you wish. Their values can be substituted in whatever case you may have. Before writing more on this line I will wait and see if this goes into the Editor's waste basket, hoping that I have made it plain enough to be easily understood by the rank and file of struggling apprentices.

\* \* \*

## SPUR GEAR ARITHMETIC.—2.

ARTHUR B. BABBITT.

PITCH (Continued).

It can be readily seen that the circular pitch being equal to the distance from the center of one tooth to the center of the next, must be the result of dividing the circumference of the pitch circle by the number of teeth in the gear. Should an occasion arise where it would be necessary to obtain the circular pitch, having the diametral pitch given, divide 3.1416 by the diametral pitch and the quotient will be the circular pitch, or, expressed in its simplest form,

$$\frac{3.1416}{P} = P^1 \quad (1)$$

in which P = diametral pitch.

$P^1$  = circular pitch.

Example: If the diametral pitch of a gear is 4 and it is required to find the circular pitch, divide 3.1416 by 4, and the quotient, .7854, is the circular pitch of the gear.

If the circular pitch be given to find the diametral pitch, we can readily see that formula (1) would have to be transposed and would read thus:

$$\frac{3.1416}{P^1} = P \quad (2)$$

P and  $P^1$  representing the same as before.

Now, having given the rules, we will proceed to explain how they were obtained. We all know that the distance around the circumference of a circle is equal to 3.1416 multiplied by the diameter of the circle, consequently for every inch of diameter we have 3.1416 inches of circumference. If the diametral pitch of a gear is equal to the number of teeth for each inch of pitch diameter, and each inch of diameter is represented by 3.1416 inches of circumference, then the diametral pitch equals number of teeth for each 3.1416 inches of circumference. As the circular pitch is the distance from the center of one tooth to the center of the next, then the circular pitch must be equal to 3.1416 divided by the number of teeth in that 3.1416 inches of circumference, and, as we have shown that the diametral pitch is equal to the number of teeth in each 3.1416 inches of circumference, then the circular pitch must equal 3.1416 divided by the diametral pitch, which proves formula 1.

It may not be actually necessary to show how we obtain the circular pitch from the diametral pitch, but as was stated at the beginning of this article, I will endeavor to explain everything as I go along. As in the preceding case, we begin with the ratio of the circumference of the circle to its diameter, which is 3.1416. In each 3.1416 inches of circumference we have a certain number of teeth which is the diametral pitch of the gear. Now, having given the diametral pitch, if we divide 3.1416 by that, we obtain the distance between the centers of two adjacent teeth, which is the circular pitch of the gear, which proves formula 2.

The following tables will facilitate the finding of corresponding diametral and circular pitches. Table No. 1 gives the even diametral pitches with the corresponding circular pitches, while Table No. 2 gives the even circular pitches with the corresponding diametral pitches:

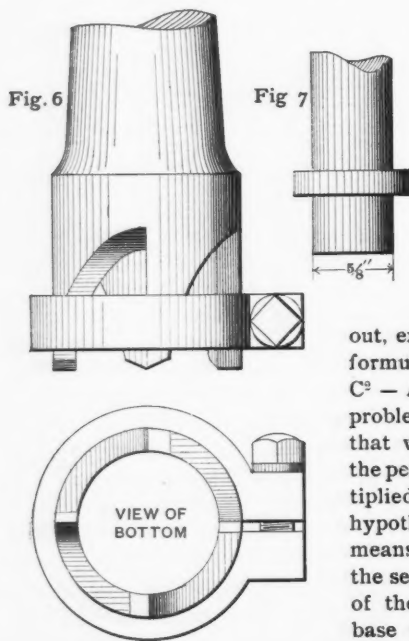


TABLE NO. 1.

Diametral Pitch.	Circular Pitch.	Diametral Pitch.	Circular Pitch.
2	1.571 inches.	12	.262 inches.
2 1/4	1.396 "	14	.224 "
2 1/2	1.257 "	16	.196 "
2 3/4	1.142 "	18	.175 "
3	1.047 "	20	.157 "
3 1/2	.898 "	22	.143 "
4	.785 "	24	.131 "
5	.628 "	26	.121 "
6	.524 "	28	.112 "
7	.449 "	30	.105 "
8	.393 "	32	.098 "
9	.349 "	36	.087 "
10	.314 "	40	.079 "
11	.286 "	48	.065 "

TABLE NO. 2.

Circular Pitch.	Diametral Pitch.	Circular Pitch.	Diametral Pitch.
2 inches.	1.571	3/8 inches.	3.590
1 7/8 "	1.676	1/2 "	3.867
1 3/4 "	1.795	5/8 "	4.189
1 1/2 "	1.933	3/4 "	4.570
1 1/4 "	2.094	7/8 "	5.027
1 1/8 "	2.185	1 "	5.585
1 1/16 "	2.285	1 1/8 "	6.283
1 1/32 "	2.394	1 1/4 "	7.181
1 1/64 "	2.513	1 1/2 "	8.378
1 1/128 "	2.646	1 3/4 "	10.053
1 1/256 "	2.793	2 "	12.566
1 1/512 "	2.957	2 1/4 "	16.755
1 1/1024 "	3.142	3 "	25.133
1 1/2048 "	3.351	4 "	50.266

## PITCH DIAMETER.

Having given the diametral pitch and number of teeth in a gear to find the pitch diameter, divide the number of teeth by the pitch and the quotient will be the pitch diameter, which, expressed in its simplest form, is:

$$\frac{N}{P} = D \quad (3)$$

in which N = number of teeth.

P = pitch (diametral).

D = pitch diameter.

Example: A 10-pitch gear has 35 teeth, what is the pitch diameter? Divide 35 (the number of teeth) by 10 (the pitch), and the quotient 3 1/2 is the pitch diameter of the gear.

The definition of diametral pitch proves this formula. If the diametral pitch equals the number of teeth to each inch of pitch diameter, then dividing the number of teeth in the gear by the diametral pitch will give the number of inches of the pitch diameter. If the circular pitch and number of teeth are given, first find the diametral pitch and proceed as given above.

## ADDENDUM.

The addendum of a gear tooth is the distance from the pitch circle to the outside circle of the gear. This distance is always equal to the reciprocal of the diametral pitch, or 1 divided by the diametral pitch, and, expressed as a formula, is:

$$A = \frac{1}{P} \quad (4)$$

in which A = addendum.

P = diametral pitch.

## OUTSIDE DIAMETER.

When we start to make a gear we first wish to know the outside diameter. If we have the pitch and number of teeth given this may easily be found by the following rule: Add 2 to the number of teeth and divide by the pitch. This, expressed as a formula, is:

$$\frac{N + 2}{P} = D^1 \quad (5)$$

in which N = number of teeth.

P = diametral pitch.

D<sup>1</sup> = outside diameter.

Example: Given a gear of 20 teeth and 4 pitch to find the outside diameter. The number of teeth, 20, plus 2 equals 22, and 22 divided by 4 (the pitch of the gear) equals 5 1/2, the outside diameter of the gear.

This formula is simply a combination of formulæ 3 and 4, for we first find the pitch diameter and then add the addendum twice, for it must be added on each side of the pitch diameter. The mathematical solution is as follows:

$$\frac{N}{P} = D \quad (3) \quad D + \frac{1}{P} + \frac{1}{P} = D^1$$

$$D^1 = D + \frac{2}{P}$$

$$D^1 = \frac{N + 2}{P} \quad (5)$$

## DEDENDUM AND CLEARANCE.

The dedendum is the working depth of the tooth below the pitch line, and must be equal to the addendum or  $\frac{1}{P}$ , for the pitch circles of two gears are tangent (touching), so the addendum of one will give the working depth of the other below the pitch line. The clearance is the distance from the end of the dedendum to the bottom of the space between the teeth. There is no common standard for this distance, different gear makers using different distances, yet the difference between them is very slight.

The Brown & Sharpe formula for this distance is:

$$F = \frac{.157}{P} \quad (6)$$

in which F = clearance.

P = diametral pitch.

The Geo. B. Grant formula is:

$$F = \frac{A}{8} \quad (7)$$

in which F = clearance.

A = addendum.

## THICKNESS OF TOOTH.

The thickness of tooth and width of the space of a gear are always equal at the pitch line, and if the circular pitch is the distance from the center of one tooth to the center of the next tooth measured on the pitch line, tooth and space being equal, then thickness of tooth must be equal to one-half the circular pitch, or

$$T = \frac{P^1}{2} \quad (8)$$

in which T = thickness of tooth at pitch line.

P<sup>1</sup> = circular pitch.

We know by formula (1) that

$$P^1 = \frac{3.1416}{P} \quad (1)$$

and substituting this value for P<sup>1</sup> in formula (8) we have:

$$T = \frac{\frac{3.1416}{P}}{2}$$

and this formula resolved to its simplest form is:

$$T = \frac{1.5708}{P} \quad (9)$$

in which T = thickness of tooth at pitch line.

P = diametral pitch.

Example: Given a gear 1 1/8 circular pitch, what is the thickness of tooth at the pitch line? 1 1/8 (the circular pitch) divided by 2 gives 11/16 the thickness of tooth at the pitch line.

Example: Given a 6 pitch gear to find the thickness of tooth at the pitch line. 1.5708 divided by 6 (the diametral pitch of the gear) gives .262 the thickness of tooth at the pitch line.

Table No. 3 gives the thickness of tooth at the pitch for the different diametral pitches:

TABLE NO. 3.

Diametral Pitch.	Thickness of Tooth at Pitch Line.	Diametral Pitch.	Thickness of Tooth at Pitch Line.
2	.785 inches.	12	.131 inches.
2 1/4	.697 "	14	.112 "
2 1/2	.628 "	16	.098 "
2 3/4	.570 "	18	.087 "
3	.523 "	20	.079 "
3 1/2	.448 "	22	.071 "
4	.393 "	24	.065 "
5	.314 "	26	.060 "
6	.262 "	28	.056 "
7	.224 "	30	.052 "
8	.196 "	32	.049 "
9	.175 "	36	.044 "
10	.157 "	40	.039 "
11	.143 "	48	.033 "

\* \* \*

Mr. JOHN V. BEEKMAN, who for thirty years has been the superintendent of the Lidgerwood Manufacturing Company, having entirely recovered from his ill-health of the last two years, has returned and taken up his duties again.

## THE HARTFORD MEETING.\*

### CURRENT PRACTICE IN ENGINE PROPORTIONS.

JOHN H. BARR, ITHACA, N. Y.

Since the presentation of the paper on "Proportions of High-Speed Engines" (given in our issue of February, 1896), a similar investigation has been made upon "low-speed" engine proportions, mainly of the Corliss type; and the original data, with some additions, have recently been revised.

No elaborate argument in justification of the use of formulas for the purpose of designing will be offered in connection with this paper. The writer is well aware of the prejudice against such instruments in certain quarters, and he has himself the most profound respect for that sound engineering judgment which often properly outweighs computations. In explanation of the predominating idea underlying this work, a quotation is made from the introduction of the paper referred to above:

"It occurred to the writer, some two or three years ago, that it might be possible to derive formulas which would express, more or less closely, the general conclusions arrived at as the result of experience in engine construction. These formulas are necessarily empirical in the sense that they are adjusted to agree with observations; but they should be, whenever possible, rational in form. That is, the variables should enter the formulas as they would enter purely analytical formulas; while the constants would be derived from practice, and not from assumed working strength, bearing pressures, etc. In other words, the engine in actual operation takes the place of the laboratory testing machine in supplying data for design.

"The advantages of using expressions of the rational form, rather than purely empirical formulas, are: first, that working stresses, factors of safety, etc., can be deduced from their constants, and that these constants can be intelligently modified to meet new conditions; second, that they can be applied with greater safety somewhat beyond the range of data from which they are obtained."

This work on low-speed engines was done by Messrs. L. J. Gray and W. S. Goll, and was presented by them as a thesis upon graduation from Sibley College, Cornell University, in 1896. Mr. T. A. Bennett of the present senior class, in the same institution, has assisted in putting the data into the shape which they now assume.

The following notation is used throughout this paper:

$D$  = diameter of piston;  $A$  = area of piston;  $L$  = length of stroke;  $S$  = steam pressure, taken at 100 pounds per square inch above exhaust, as a standard pressure; HP. = rated horsepower;  $N$  = revolutions per minute;  $C$  = a constant. All dimensions in inches, unless stated to the contrary. Other notation is explained as used.

**Crank Shaft.**— $d$  = diameter of shaft. The formula for the diameter of a shaft which is subjected to torsion is

$$d = C \sqrt[3]{\text{HP.} \div N}, \text{ if the moment of torsion is constant.}$$

Crank shafts are subject to variable combined bending and twisting moments; but these moments, when their magnitude and variation are known, can be reduced to an equivalent twisting moment; hence an expression of the above form applies to the case in hand, if the ratios between bending to twisting moments and between maximum and mean moments are constant. These ratios should not affect the form of the above expression, but only the value of the numerical coefficient. In the engines examined, there is a general agreement as to the above ratios of moments among the engines of the same class. Of course this agreement is by no means mathematically exact; but the constants given in this paper are only intended to show the general trend of practice, and the diagrams exhibit the uniformity, or lack of uniformity, among the various builders as to certain proportions.

#### THE CYLINDER.

Data on thickness of cylinder walls (shell), flanges, cylinder heads, and cylinder head bolts were obtained only for the engines classed as low-speed.

**Thickness of Walls.**—The shell of the cylinder must have sufficient thickness to resist the maximum bursting action and to avoid objectionable deformation, even after the cylinder has been reboiled one or more times; and in small cylinders, for moderate pressures, the thickness necessary to insure good castings may be the prime requirement. Such considerations have

led to the proposal of empirical formulas in which the steam pressure does not appear. While the use of this class of formulas is not in strict accord with a leading idea of the work here described, it has seemed well to adopt such an expression in this instance.

The general form of the expression that is very often employed:

$$t = CD + B,$$

in which  $t$  = the thickness of the shell in inches,  $D$  = the diameter of the piston in inches, and  $C$  and  $B$  are the constants. From the engines examined it was found that  $C$  varies from .04 to .06, and that  $B$  = .3 inches.

The general practice is expressed approximately by

$$t = .05 D + .3 \text{ inches.}$$

It seems not unreasonable to look upon the added constant of 0.3 inch, found above, as an allowance for reboiling, etc., and the coefficient  $C$  as one which should vary with the steam pressure. The engines considered in deriving the values given are all rated on from 80 to 100 pounds pressure; hence, in using this formula for higher pressures it would seem advisable to increase the value of  $C$  proportionally. Looked at from this standpoint, the above formula becomes a rational one.

**Flanges and Cylinder Heads.**—The flanges and the cylinder heads usually have the same general thickness. This is found to vary from 1.0 to 1.5 times the thickness of the shell, the mean value being about 1.2 times the thickness of the cylinder wall.

**Cylinder Head Studs.**—There is no general agreement as to the number of studs nor their diameters. In the above specified low-speed engines none have studs less than  $\frac{3}{4}$  inch or more than  $1\frac{3}{4}$  inches diameter. The least number of bolts is 8 for a cylinder 10 inches diameter, and the greatest number is 32 for a cylinder 24 inches diameter.

A large number of small bolts, as against a small number of large bolts, tends to secure tightness of the joint; but the smaller bolts are subjected to greater stress in screwing up. It may be mentioned in this connection that experiments, made under the direction of the writer, show that the stress at the bottom of the thread, due to screwing up a bolt, may equal or exceed  $\frac{30,000}{d}$  pounds per square inch, in which  $d$  = the nominal diameter of the bolt.

The average number of bolts used in each head of the above engines is given approximately by

$$n = .7 D,$$

in which  $n$  = number of bolts, and  $D$  = diameter of piston in inches. Of course the number given by this rule would usually be modified to secure an even number.

The general practice as to diameter of studs is represented nearly by

$$d = \frac{D}{40} + \frac{1}{2} \text{ inches,}$$

$d$  being the nominal diameter of the studs.

#### PORTS AND PIPES.

**Areas of Ports and Pipes.**—

Area of port (or pipe) =  $a$ , in square inches.

Area of piston =  $A$ , in square inches.

Mean piston speed =  $V$ , in feet per minute.

The relation of port area (or pipe area) to area of piston and mean piston velocity is expressed by

$$a = \frac{AV}{C},$$

in which  $C$  is the mean velocity of steam through the port, or pipe, in feet per minute.

**Ports—High-Speed Engines.**—(The same ports used for steam and exhaust.)

For the general practice it is found that

Mean value of  $C$  = 500. Maximum value of  $C$  = 650.

Minimum value of  $C$  = 450.

As the piston speed is quite constant for a large number of these engines (about 600 feet per minute), the area of port may be conveniently expressed by

$$a = KA,$$

in which  $K$  is as follows for the general practice:

Mean value of  $K$  = .10. Maximum value of  $K$  = .13.

Minimum value of  $K$  = .07.

\* Of the A. S. M. E.

*Area of Steam Ports—Low-Speed Engines.*—(Separate ports for exhaust.)

For these engines it is found that the general practice is represented by

Mean value of  $C = 6800$ . Maximum value of  $C = 9000$ .  
Minimum value of  $C = 5000$ .

In the relation  $a = KA$ ,  $K$  varies for the general practice with these engines as follows:

Mean value of  $K = .09$ . Maximum value of  $K = .10$ .  
Minimum value of  $K = .08$ .

*Exhaust Ports—Low-Speed Engines.*—With the same forms of expressions as above, designating area of the exhaust port by  $a$ , it is found that

Mean value of  $C = 5500$ . Mean value of  $K = .11 = \frac{1}{9}$ .  
Maximum value of  $C = 7000$ . Maximum value of  $K = .125 = \frac{1}{8}$ .  
Minimum value of  $C = 4000$ . Minimum value of  $K = .10 = \frac{1}{10}$ .

*Steam Pipes—High-Speed Engines.*—In the expression

$$a = \frac{AV}{C},$$

Mean value of  $C = 6500$ . Maximum value of  $C = 7000$ .  
Minimum value of  $C = 5800$ .

As the piston speed is approximately the same in many of the cases, it is convenient to use the relation

$$d = KD.$$

It is found that

Mean value of  $K = .35$ . Maximum value of  $K = .40$ .  
Minimum value of  $K = .30$ .

*Steam Pipes—Low-Speed Engines.*—With these engines it is found that

Mean value of  $C = 6000$ . Mean value of  $K = .32$ .  
Maximum value of  $C = 8000$ . Maximum value of  $K = .38$ .  
Minimum value of  $C = 5000$ . Minimum value of  $K = .26$ .

*Exhaust Pipes—High-Speed Engines.*—

Mean value of  $C = 4400$ . Mean value of  $K = .40$ .  
Maximum value of  $C = 5500$ . Maximum value of  $K = .50$ .  
Minimum value of  $C = 2500$ . Minimum value of  $K = .35$ .

*Exhaust Pipes—Low-Speed Engines.*—

Mean value of  $C = 3800$ . Mean value of  $K = .4$ .  
Maximum value of  $C = 4700$ . Maximum value of  $K = .45$ .  
Minimum value of  $C = 2800$ . Minimum value of  $K = .35$ .

#### FACE OF PISTONS.

It is not to be expected that any very general agreement will be found in the practice of various builders as to the face, or width, of pistons.

\* The following expression, in which  $F$  = face of piston, was employed to express the relation of face of piston to diameter of piston:

$$F = CD.$$

*High-Speed Engines.*—

Mean value of  $C = .46$ . Maximum value of  $C = .60$ .  
Minimum value of  $C = .30$ .

*Low-Speed Engines.*—

Mean value of  $C = .32$ . Maximum value of  $C = .45$ .  
Minimum value of  $C = .25$ .

No data were obtained on the thickness of piston walls.

#### PISTON RODS.

There are several methods of treating this member analytically, any of which might be taken as the basis of a formula in deriving constants by the general method used in this investigation.

It has seemed best to treat it as a long strut, to be designed for rigidity, inasmuch as any considerable buckling or flexure of the rod would induce objectionable friction and wear at the stuffing box, or possibly cramping of the piston. The Euler formula has been followed. It has the form

$$P = K \frac{EI}{L^2}; \quad P = K\pi^2 \frac{EI}{L^2},$$

in which  $E$  = the modulus of elasticity;  $I$  = moment of inertia for the section;  $L_1$  = the length of the strut; and  $P$  = the greatest load consistent with stability. The value of the constant  $K$  depends upon the end conditions; that is, upon whether the ends of the strut are "fixed" or pivoted, free or guided. The piston rod is considered as coming under the case in which the strut is fixed at one end and free at the other. It may be urged that the guides constrain the outer end; but many forms

\* Only horizontal engines are included in deriving the following coefficients.

of guides are poorly adapted to exert constraint against lateral flexure, and it is better to provide the slight increase of diameter required to avoid such side pressure on the guides. In this case

$$K = \frac{1}{4},$$

or

$$P = \frac{\pi^2 EI}{4 L_1^2}.$$

$$P = S \frac{\pi}{4} D^2; \quad I = \frac{1}{64} \pi d^4; \quad L_1 \text{ (in plotting) is taken equal to}$$

length of stroke  $L$ . Of course the free length of a piston rod is always somewhat greater than the length of stroke, but their ratio is not very different in most engines of a similar class; hence we may use  $L$  for  $L_1$ , with proper modification of the constant involved in the formula.

General practice will be assumed to show that the free length of piston rod  $L_1 = 1.2 L$  for high-speed engines, or that  $L_1^2 = 1.4 L^2$ ; and that  $L_1 = 1.1 L$  for low-speed engines, or  $L_1^2 = 1.2 L^2$ .

Assuming  $S$  as a standard steam pressure of 100 pounds per square inch, and that  $E = 30,000,000$ , the Euler expression takes the form for high-speed engines (with  $L_1^2 = 1.4 L^2$ ):

$$\frac{\pi D^2}{100} = \frac{\pi^2 \times 30,000,000 \pi d^4}{4 \times 1.4 L^2 \times 64},$$

or

$$d^4 = \frac{64 \times 100 \times 1.4}{30,000,000 \pi^2} (D^2 L^2);$$

therefore  $d = .074 \sqrt{DL}$

In a similar way for the low-speed engines:

$$d = .07 \sqrt{DL} \text{ (taking } L_1^2 = 1.2 L^2 \text{)}.$$

Both of these expressions are for a factor of safety of unity.

*Piston Rods of High-Speed Engines.*—

Mean value of  $C = .145$ . Maximum value of  $C = .175$ .  
Minimum value of  $C = .12$ .

It will be seen that this mean value of  $C$  will give a diameter of rod greater than that required with a factor of safety of unity in the ratio of .145 to .074, or of 2 to 1 (nearly). As the strength of the long strut varies as the fourth power of the diameter, the

factor of safety is  $\left(\frac{.145}{.074}\right)^4 = 15$ .

*Piston Rods of Low-Speed Engines.*—

Mean value of  $C = .11$ . Maximum value of  $C = .13$ .  
Minimum value of  $C = .10$ .

The diameter given by this mean value of  $C$  is to the diameter with factor of safety of unity as .11 to .07, or as 1.57 to 1. This corresponds to a factor of safety of  $\left(\frac{.11}{.07}\right)^4 = 6$ .

While the conditions under which a high-speed engine operates may make a higher factor of safety proper with this type, it may be questioned whether so great a difference is really necessary. Two elements, not yet mentioned, should be considered in this connection, however. The body of the rod should be not only stiff enough as a long strut, but the ends, where they are reduced by screw threads or key-ways, must have a section sufficient for the direct stress (tension and compression). The area of these reduced sections should be about the same for engines of the same diameter (with similar materials and steam pressure), regardless of the length of stroke; hence the diameter of body of the rod which provides the required reduced area will give a greater margin with the shorter strokes. Furthermore—

Since the strength of the long strut increases as the fourth power of the diameter, it requires an increase of diameter of less than 20 per cent. to double the factor of safety; therefore the tendency to provide somewhat greater dimensions with high-speed engines than are common in low-speed engines of similar diameter of piston may easily lead to an excessive increase in the factor of safety in such a member as the piston rod.

#### CONNECTING RODS.

These are treated, like the piston rods, as long struts, except that the constants are different, owing to the end conditions. The connecting rod is "pin ended," or round ended, as regards buckling in the plane of its motion, while it is to be treated as "square ended" against lateral deflection. Rods of circular cross section need only be considered with reference to the

plane in which they are most liable to buckle; viz., in the plane of their motion. Rods of rectangular section (or of approximately rectangular section) are liable to buckle in either plane, depending upon the relation of the height,  $h$ , to the breadth (thickness),  $b$ , of the section.

The Euler formula for a pin-ended strut is

$$P = \pi^2 \frac{EI}{L_1^2},$$

and for a square-ended strut it is

$$P = 4\pi^2 \frac{EI}{L_1^2},$$

in which  $P$  is the greatest load consistent with stability;  $E$  is the modulus of elasticity;  $I$  is the moment of inertia of the mid-section; and  $L_1$  is the length of the struts.

*Connecting Rods of High-Speed Engines.*—(Rectangular sections only.)

Connecting rods generally six cranks long.

It can be shown from the above expressions that a rectangular strut with end conditions as in connecting rods, under static load, should have a cross section such that the dimensions in the plane of the pin axes is one-half of the dimension perpendicular to this plane, or  $h = 2b$ . In engine connecting rods, the value of  $h$  is considerably greater than  $2b$ . This excess of  $h$  over  $2b$  may be considered as an allowance for the inertia action (centrifugal throw) on the rod; that is, we may consider the strength of the rod against the thrust as determined by the breadth  $b$  and a height  $2b$ . Then, for lateral flexure,

$$I = \frac{1}{12} b^3 h = \frac{1}{12} b^3 \times 2b = \frac{1}{6} b^4.$$

With a load  $P = \frac{1}{4} \pi D^2 S$  (in which  $S$  is taken at 100 pounds per square inch), a factor of safety of unity, and  $E = 30\,000\,000$ :

$$\frac{1}{4} \pi D^2 \times 100 = 4\pi^2 \times 30\,000\,000 \times \frac{b^4}{6L_1^2}.$$

Therefore  $b = .025 \sqrt{DL_1}$ .

Adopting the expression

$$b = C \sqrt{DL_1}$$

for the connecting rods, plotting  $b$  and  $\sqrt{DL_1}$  of each engine as coordinates, and drawing lines to represent the mean, maximum, and minimum of the general practice, the following values of  $C$  are obtained:

Mean value of  $C = .057$ . Maximum value of  $C = .07$ .  
Minimum value of  $C = .045$ .

This mean value of  $C$  indicates a factor of safety of  $\left(\frac{.057}{.025}\right)^4 = 25$ .

A comparison of the height,  $h$ , and breadth,  $b$ , at mid-section of the above rods shows that in the relation  $h = Kb$ ,

Mean value of  $K = 2.7$ . Maximum value of  $K = 4.0$ .  
Minimum value of  $K = 2.2$ .

*Connecting Rods of Low-Speed Engines.*—(Circular sections only.)

Connecting rods generally five and a half cranks long.  
For flexure in the plane of motion

$$\frac{1}{4} \pi D^2 S = \frac{1}{4} \pi D^2 \times 100 = \pi^2 \times 30\,000\,000 \times \frac{\pi d^4}{64 L_1^2}.$$

Therefore  $d = .048 \sqrt{DL_1}$  (for factor of safety of unity).

Using the general expression

$$d = C \sqrt{DL_1},$$

plotting  $d$  and  $\sqrt{DL_1}$  as the coordinates, and drawing mean and extreme lines as before:

Mean value of  $C = .092$ . Maximum value of  $C = .105$ .  
Minimum value of  $C = .082$ .

The mean value of  $C$  indicates a factor of safety (neglecting the whipping of the rod) of

$$\left(\frac{.092}{.048}\right)^4 = 13.$$

#### CROSSHEAD SHOES.

The area of the crosshead shoes which sustains the vertical component of the force of the connecting rod is expressed in relation to the area of the piston by

$$a = CA.$$

The maximum pressure per square inch of shoe,  $p$  (if the

steam follows up for half stroke), is nearly equal to  $SA + na$ , in which  $n$  is the ratio of connecting rod length to the crank. Hence, with  $S = 100$  pounds per square inch:

$$p = \frac{100A}{na} = \frac{100A}{nC} = \frac{100}{nC} \text{ (nearly).}$$

*Crosshead Shoes of High-Speed Engines.*—

$$p = \frac{100}{6C} = \frac{17}{C},$$

Mean value of  $C = .63$ .

Mean value of  $p = 27$ .

Maximum value of  $C = 1.60$ .

Maximum value of  $p = 38$ .

Minimum value of  $C = .45$ .

Minimum value of  $p = 10.5$ .

*Crosshead Shoes of Low-Speed Engines.*—

$$p = \frac{100}{5.5C} = \frac{18.5}{C}$$

Mean value of  $C = .46$ .

Mean value of  $p = 40$ .

Maximum value of  $C = .64$ .

Maximum value of  $p = 58$ .

Minimum value of  $C = .32$ .

Minimum value of  $p = 29$ .

#### CROSSHEAD PINS.

The relation of the projected area of crosshead pin to area of piston is

$$a = (dl) = CA.$$

The ratio of diameter to length of crosshead pin is expressed by  $l = Kd$ .

*Crosshead Pins of High-Speed Engines.*—

Mean value of  $C = .08$ .

Mean value of  $K = 1.25$ .

Maximum value of  $C = .11$ .

Maximum value of  $K = 2.0$ .

Minimum value of  $C = .06$ .

Minimum value of  $K = 1.0$ .

*Crosshead Pins of Low-Speed Engines.*—

Mean value of  $C = .07$ .

Mean value of  $K = 1.3$ .

Maximum value of  $C = .10$ .

Maximum value of  $K = 1.5$ .

Minimum value of  $C = .054$ .

Minimum value of  $K = 1.0$ .

#### CRANK PIN.

The lost work per minute due to friction of a journal may be expressed by  $Pu\pi dN$ , in which  $P$  = mean pressure on the journal;  $d$  = diameter of the journal;  $N$  = revolutions per minute; and  $u$  = the coefficient of friction.

The heat resulting from this frictional work is dissipated through a surface which is proportional to the projected area of the journal ( $dl$ ), and reason and experience indicate that this projected area should be about proportional to the heat so dissipated.

Therefore

$$\frac{Pu\pi dN}{dl} = \frac{Pu\pi N}{l} = K.$$

Therefore

$$l = \frac{Pu\pi N}{K}.$$

In an engine, if  $P$  = the mean total pressure on piston,

$$P = \frac{12 \times 33\,000 \text{ HP.}}{2LN}.$$

Therefore

$$l = \frac{12 \times 33\,000 \times u\pi \text{ HP.}}{2KL} = C \frac{\text{HP.}}{L}.$$

Upon plotting the data, it was found that the length is expressed better by the form

$$l = C \frac{\text{HP.}}{L} + B$$

(in which  $B$  is an added constant) than by the above rational form.

Projected area of crank pin:  $(dl) = KA$ .

*Crank Pins of High Speed Engines.*—

Mean value of  $C = .30$ ; of  $B = 2.5$  inches.

Maximum value of  $C = .46$ ; of  $B = 2.5$  inches.

Minimum value of  $C = .13$ ; of  $B = 2.5$  inches.

Mean value of  $K = .22$ .

Maximum value of  $K = .40$ .

Minimum value of  $K = .10$ .

*Crank Pins of Low Speed Engines.*—

Mean value of  $C = .6$ ; of  $B = 2$  inches.

Maximum value of  $C = .8$ ; of  $B = 2$  inches.

Minimum value of  $C = .4$ ; of  $B = 2$  inches.

Mean value of  $K = .09$ .

Maximum value of  $K = .115$ .

Minimum value of  $K = .065$ .

## CRANK SHAFTS, MAIN JOURNALS.

The method of treating the diameter of the crank shaft at main journal was referred to in the introductory portion of this paper. The expression used is

$$d = C \sqrt[3]{\frac{HP}{N}}$$

The ratio of length to diameter of main journal is expressed by  $l = Kd$ .

The projected area of the main journal is given in terms of the piston area by  $dI = MA$ .

*Main Journals of High-Speed Engines\*.*

Mean value of $C = 7.3$ .	Mean value of $K = 2.2$ .
Maximum value of $C = 8.5$ .	Maximum value of $K = 3.0$ .
Minimum value of $C = 6.5$ .	Minimum value of $K = 2.0$ .
Mean value of $M = .46$ .	Maximum value of $M = .70$ .
Minimum value of $M = .37$ .	

*Main Journals of Low-Speed Engines†.*

Mean value of $C = 6.8$ .	Mean value of $K = 1.9$ .
Maximum value of $C = 8.0$ .	Maximum value of $K = 2.1$ .
Minimum value of $C = 6.0$ .	Minimum value of $K = 1.7$ .
Mean value of $M = .56$ .	Maximum value of $M = .64$ .
Minimum value of $M = .46$ .	

## PISTON SPEED.

The mean piston speed, or the piston travel in feet per minute, is

$$V = \frac{2LN}{12} = \frac{LN}{6}$$

*High-Speed Engines.*

Mean value of $V = 600$ .	Maximum value of $V = 660$ .
Minimum value of $V = 530$ .	

*Low-Speed Engines.*

Mean value of $V = 600$ .	Maximum value of $V = 850$ .
Minimum value of $V = 500$ .	

## WEIGHT OF RECIPROCATING PARTS.

The weights of piston, piston rod, crosshead and connecting rod were obtained. These weights are plotted only for the high-speed engines.

For engines having similar compression, smoothness of running (in passing the dead points) indicates that the weight of reciprocating parts,

$W$ , should be proportional to  $\frac{D^3}{LN^2}$ . The reciprocating parts are taken as the piston, piston rod, crosshead, and one-half the connecting rod.

$$W = C \frac{D^3}{LN^2}$$

Mean value of $C = 1\ 860\ 000$ .
Maximum value of $C = 2\ 300\ 000$ .
Minimum value of $C = 1\ 200\ 000$ .

## BELT SURFACE PER I. H. P.

The relation of the belt width and velocity to the indicated horse-power is expressed by  $S = C$  (HP.), in which  $S$  is the product of the width of belt in feet multiplied by the velocity of the belt in feet per minute.

*High Speed Engines.*

Mean value of $C = 55$ .	Maximum value of $C = 70$ .
Minimum value of $C = 40$ .	

*Low Speed Engines.*

Mean value of $C = 35$ .	Maximum value of $C = 42$ .
Minimum value of $C = 30$ .	

## FLY WHEEL.

Data as to the flywheels were obtained for the high-speed engines only.

The weight of rim is expressed by

$$W = \frac{C HP}{D_1^2 N^3}$$

in which  $W$  = weight of the rim in pounds, and  $D_1$  = diameter of the wheel in inches.

Mean value of $C = 1\ 700\ 000\ 000\ 000$ .
Maximum value of $C = 2\ 000\ 000\ 000\ 000$ .
Minimum value of $C = 650\ 000\ 000\ 000$ .

\* These values are for each of the two journals of center-crank engines.  
† One journal only: side-crank engines.

The velocity of rim in these engines has a general value of about 4 200 feet per minute, or 70 feet per second.

## WEIGHT OF ENGINE PER I. H. P.

The weight of engine (including flywheel) per I. H. P. is  $W = CH - P$ .

*High-Speed Engines.*

Mean value of $C = 115$ .	Maximum value of $C = 135$ .
Minimum value of $C = 100$ .	

*Low-Speed Engines.*

Mean value of $C = 175$ .	Maximum value of $C = 240$ .
Minimum value of $C = 135$ .	

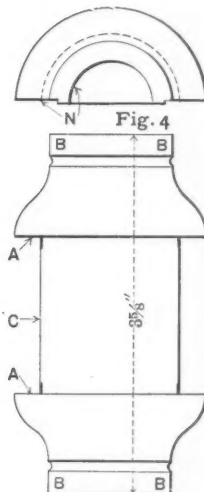
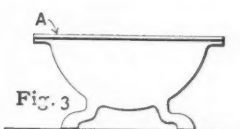
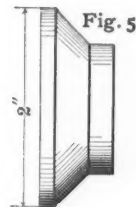
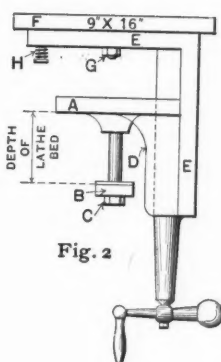
\* \* \*

## SEEN IN VARIOUS SHOPS.—3.

## A. HARD CASE

Fig. 2 is a fixture that the writer saw at the Diamond Machine Co., of Providence, R. I. A is a plate that fits the lathe bed exactly like a center rest and fastened by the strap B and nut C, with D projecting down inside of the lathe bed. E is fitted to this by the usual dovetail groove and gib, and bends over A as shown. A larger table F for such work as requires it is pivoted to this by the stud G. H is a cone pointed spring catch that locks the table F at each quarter of a turn, but not so solidly but what it will slip out when a reasonable amount of force is applied to the movable table F. Either table E or F can be raised to the level of the lathe centers or lowered by the ball crank and screw. Several of these "rigs" were provided for the shop, and the workmen seemed to appreciate them. Pulleys or other work placed upon the table for drilling, etc., was raised to the proper height and in this way many ordinary jobs were easily drilled sufficiently uniform and accurate, or even better than I have seen similar work done by some men after spending a considerable time laying each piece out separately.

Near one of these used for drilling pulleys, sat an old grindstone trough, Fig. 3, which had been planed off true on top, leaving two narrow raised ribs A for balancing pulleys on in the



ordinary way, instead of being put into the cupola, as "A Roving Contributor" would advise for one of this design.

Another thing noticed was a lot of 800 small bench grinders under way, and instead of making the bearings by planing the two parts together and then boring for the spindle, proper fixtures were provided for holding them, and a gang of formed milling cutters used, producing good results quickly and leaving only enough in the hole for hand reaming.

Fig. 4 is a cap for a small bearing, and the entire under side A was milled at one cut as noted. The next operation was to clamp them to an arbor in pairs by two rings with screws bearing at B B B, and A C A finished to gauge by set tools. Then they were next held by this part in a special fixture in the same manner as when in use, and one end finished on a screw machine in two cuts with formed tools. Two more cuts finished the other end, making it an exact duplicate of the first. I suppose it is needless to call attention to the extra cost of such pieces as made in some shops, caused by a little unnecessary difference in the curves on the two ends.

I was shown both halves of a larger box of this style (1 1/2 inch shaft), taken from a grinder built by this company, after four years' service in a foundry, evidently without "roughing up" or

being adjusted. It was cast iron and perfectly glazed, seeming more like hardened steel than anything else. No ring or roughness was visible, although it was worn about  $\frac{3}{8}$  of an inch.

Three thousand two hundred collars similar to Fig. 5 was another screw machine job. These were drilled and finished all over at one setting, except on the face, which would have to be finished after being forced on the spindle any way.

Mr. Alvah Wiley, one of the foremen, is evidently an enthusiastic supporter of the milling machine for his class of work, but his own words are best to express his opinion of the Jones & Lampson flat turret lathe, and are worth repeating. "If a man wishes to reduce costs, that is what he wants to get to do it with. You see we use regular lathe tools there, and instead of setting one and making the rest project far enough to come up and stop at the right place like most turret lathes, each tool is gauged to stop at the desired point independently by those back stops."

Besides their regular lines of well known grinding machinery, this company is adding some new designs and building a number of special machines, among them are machines for grinding helical work and others for grinding the ball races used on bicycles, etc., yet this company does not seem afraid for any one interested to see the details or the quality of the work turned out by their methods.

\* \* \*

### VALVE GEARS.—9.

E. T. ADAMS.

#### MOTION CURVES.

It has already been shown that the rate of opening and closing of the valves is an important factor in determining the form of card that will be obtained from a given valve gear. Something of this may be determined from the Bilgram diagram, but after all there is nothing which can wholly take the place of the motion curves or curves of port opening, which show at a glance not only the actual opening at any point in the stroke, but also the rate at which this opening increases or decreases. There are several methods of drawing such curves, but perhaps the simplest, and for our purpose the best, is drawn as follows:

Draw the line A B, Fig. 36, any convenient length, say four inches; this represents the stroke of the piston. Next, divide A B into any number of parts; each will represent some piston position. Taking a radius equal to the length of the connecting rod and striking arcs, cutting the crank-pin circle as shown, we may locate the corresponding crank-pin positions—of course the diameter of the crank-pin circle must equal A B. Now the perpendicular from Q to each of these crank positions is the corresponding displacement of the valve from mid-position, and laying off these distances at proper points, above A B for the forward stroke and below A B for the return stroke, we obtain a series of points through which we may draw the irregular ellipse shown in the figure. This ellipse is a motion curve, and a perpendicular drawn from any point in A B to this curve gives the displacement of the valve when the piston is at that point; then,

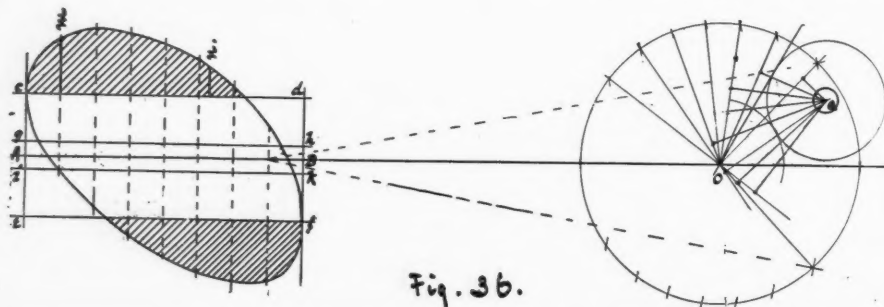


Fig. 36.

if we draw two lines, as  $cd$  and  $ef$ , parallel to A B and distant from it by the lap, these lines will cut the curve at the piston positions for admission and cut-off. Similarly two lines  $gh$  and  $ik$  distant from A B by the exhaust lap will cut this curve at the points of release and compression. When the displacement of the valve is greater than the lap, the steam port is open; hence the shaded portion of the space under the curve shows the change in the amount of port opening from admission to cut-off. Now lay off equal distances, say one-half inch, from the points of admission and cut-off and erect the two perpendiculars  $m$  and  $n$ . Each represents the port opening at that instant. It is evident at once that the port opening is considerably greater when

the piston has advanced a certain distance beyond its admission position than it is when the piston lacks the same distance of reaching the point of cut-off. Of course the valve is moving with the same velocity in each case, but the piston is moving much faster at cut-off than it is at admission, and when cut-off occurs at a point anywhere near mid-stroke the result is a somewhat sluggish closing of the valve, which is indicated by the rounded corner of the card at cut-off. If the steam and exhaust laps are unequal at the two ends of the valve, as they will be if the valve is designed and set for equal cut-off, the lines  $g'h'$  and  $i'k'$ , etc., should be drawn at the proper distances from A B. In the case chosen in the figure the valve is supposed to be designed and set for equal lead, and the inequality in the distances traversed by the piston up to the point of cut-off is very clearly

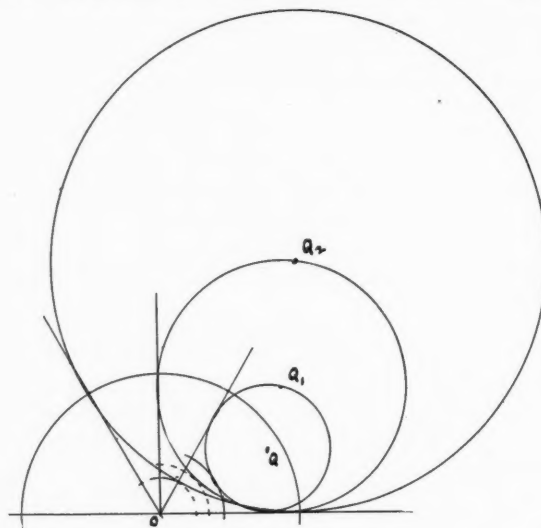


Fig. 37.

shown. The connecting rod is here assumed to be six times the length of the crank. A comparison of indicator cards with a motion curve drawn from data secured by measuring the valves of an engine in actual use will be found both interesting and valuable to any one who will undertake the task. Experiments such as this, or the study of indicator cards taken from engines with different kinds of valve gear, furnish the connecting link between theory and practice, and one who would be expert in designing valves must acquire in some such way the judgment which will enable him to predict from the form and proportions of the diagram the probable form of card which will be obtained in a given case. This is the vital part of valve gear design. It is not to be learned from books, but each man must work it out for himself by some such process as that indicated.

#### PORT OPENING AT EARLY POINTS OF CUT-OFF.

Any one who will compute the port opening required for a given engine, and will then determine the lap necessary to secure that opening when cut-off is to occur at one-quarter stroke, will readily admit that there are grave practical difficulties in the way of securing full port opening at early points of cut-off when using a plain slide valve driven by a single fixed eccentric. It is probable, however, that when cut-off occurs early in the stroke the area of port opening given by the formula is greater than necessary. As has been already explained the formula simply gives the area which has been found necessary under usual conditions; that is, when cut-off occurs late in the stroke. There is no generally accepted rule for determining the port opening necessary when cut-off occurs at early points in the stroke. Designers of Corliss, single valve automatic, or other types of engines which are generally operated at short cut-off have usually secured as much port opening as they could under the circumstances, and a comparison of indicator cards shows that there is no general agreement in the results obtained. Some guide to the area of port opening necessary at early points of cut-off may be obtained as follows: In the formula this area is made directly proportional to the average velocity of the piston. If we now determine the average velocity of the piston up to three-quarter stroke

and call this the unit velocity, then it will be found that the average velocity of the piston up to half stroke is about eight-tenths of this, and that the average velocity of the piston up to one-quarter stroke is only six-tenths of the average up to three-quarter stroke, and since the port opening is directly proportional to this velocity, it seems reasonable to assume that for any point of cut-off the port opening should be made proportional to the average velocity of the piston up to that point; that is, if the formula gives 10 square inches as the port opening, then eight-tenths of this, or 8 square inches, will be sufficient when cut-off occurs at half stroke, and six-tenths of this, or 6 square inches, will be ample when cut-off is to occur at one-quarter stroke. For points of cut-off earlier than one-quarter stroke insufficient port opening will probably be an advantage rather than otherwise and these very early points of cut off need not be considered. Fig. 37 shows the change in lap and throw of eccentric necessary to secure this theoretical port opening at three-quarter, one-half and one-quarter cut-off, when using a plain slide valve driven by a single fixed eccentric. It is evident from the figure that the size of the valve will increase very rapidly as we make cut-off occur earlier in the stroke, and in fact we soon reach a point when the friction size and weight of the valve become so great as to be practically prohibitive. Various expedients such as multi-ported valves, balancing, etc., have been employed to secure an early cut-off, but each involves the disuse or at least the radical modification of the plain slide valve and generally some change from the single fixed eccentric as well, and this is properly another story. There seems to be a tendency, in this country at least, to speak somewhat contemptuously of the plain slide valve engine and to associate it with backwoods practice generally. This is unfair. Of course this engine has its limitations, but within the range to which it is adapted, that is, at rather late cut-off, it is, when properly proportioned for its work, about as economical as any engine would be, and considering the total cost of power it will often be found more economical than many if not all of its more aristocratic competitors.

\* \* \*

#### GRAPHIC SYSTEM OF COMPUTATION.

"CISNARF."

It has always been an odd circumstance that graphic methods of computation have been laid aside from the machinery branch of engineering. Even in the beginning of a course of study, the word "*Graphics*," printed in large black letters, is handled in a manner that strikes terror to the student's heart, and he shuns it at every opportunity, as he would any other superfluity which he thinks is added as an ornamental finish to his education. We cannot blame him, as little importance is given to it by the average instructors, who, as a rule, are not familiar with the smaller details of commercial business (nor can we expect that they should be.) There are two ways of discovering the importance and application of labor and time-saving devices: one

tainly is, along with the calculating machines, fountain pens, etc., etc. We have the slide rule, a grand thing after "learning how," and by the way, when you have it with you; but "graphics," that terrible thing, calculating with lines and angles, is so very simple that any one with a little of the right sort of training and a little active matter in his brain, can make his own systems for his special work, and save time, labor and mistakes, the latter the worst part of a mechanic's life, and which, like a spectre, is ever before him during and after his work. Problems embodying the computation of levers are very nicely adapted to this method, and I give in this article *one* (there are others) which was thrust upon me as a rush order. Fig. 1 shows a portion of a sheet in which a series of holes was to be made by a reciprocating punch, (operating in a fixed slide) not in a straight line. The holes were to be made on the curve CD; the

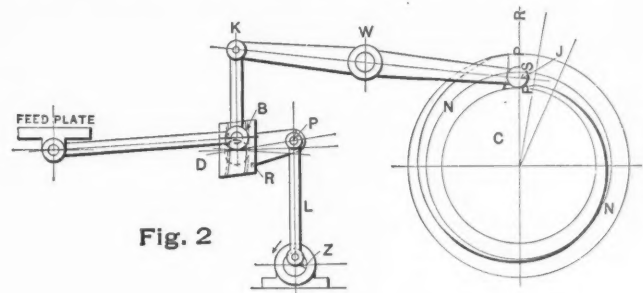
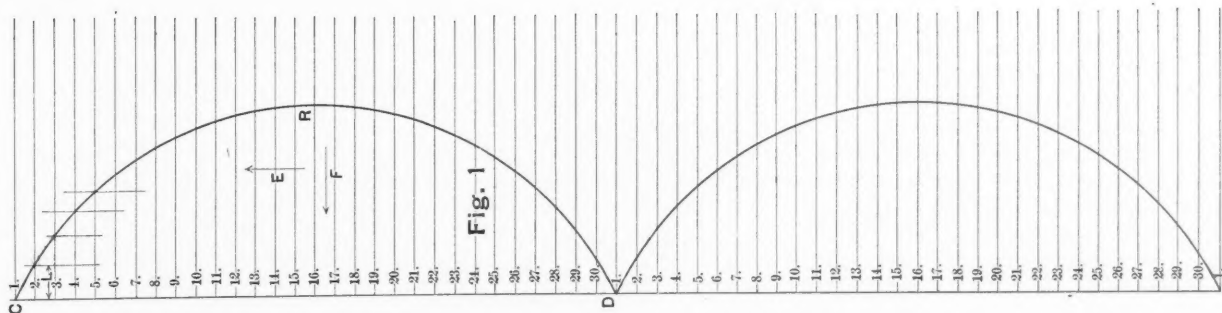


Fig. 2

feed in the direction E was constant, but in F variable. To follow the line of the curve it required 31 strokes to complete a part, when a sharp reverse occurred, beginning the next curve (it will be noticed that an easy reverse occurred at R, the sixteenth hole). The mechanism employed is shown in Fig. 2, the E feed being omitted. There is a slotted rocker R, operated through the link L by the crank Z; in this rocker R is a sliding block B, which is connected with the feed plate by the long link. This plate, during the feed, was lifted up against the under side of the sheet, but on the backward stroke, was lowered to clear it; the amount of movement of the sheet is plainly seen to be regulated by the position of B. From B is hung a link to the lever, KWJ, this in turn working on the rim of a cam C. The problem was to find the outline of this cam to regulate B, in a manner to feed in the given curve. A few minutes' calculation showed the impracticability of that method. Exactness was required, and this prohibited much dropping of decimals, which soon became too heavy to handle. Thirty different calculations had to be performed: fifteen might do, as both halves of the curve were alike, but thirty points on that cam had to be accurately located, and what was worse, in a very short space of time; five more were to be similarly treated, for different curves. First, to enable measuring, the curve to be punched was laid out six times its actual size, and treated as in Fig. 1, the holes numbered, and the E



by continual thought and search for their uses; the other by the sudden and unforeseen appearance of problems needing immediate and accurate solution. It was one of the latter that drove me from "figures" to "lines."

In every business "except ours," various things are used to economize; to be sure, they are creations of our line of business, that is, mechanical combinations; but the ways of humanity seem to be that the man mostly in need of outsiders' help to assist him in improving his system of business, is the very one who is occupied in performing this same operation for others. Shorthand, a development of geometrical principles, might be called a creation of a branch of engineering; the typewriter cer-

feed being measured from hole to hole, on Fig. 2 the throw of P was  $\frac{1}{4}$  inch; the arm PD 1 inch. A double angle was drawn, Fig. 3, on the large size adopted for the curve; OA was six times 1 inch, and AB six times  $\frac{1}{4}$  inch, or  $1\frac{1}{2}$  inch, divided equally on each side of the center line. A number of vertical lines were drawn on which to lay off distances taken from Fig. 1. At the beginning the feed L, Fig. 1, was measured with dividers, and was spaced equally as shown in Fig. 3. Then the T square was used to draw XX, cutting the oblique line on both sides of O. The vertex, the distance OX, was the height of the center of the sliding block above the center D, to give the required stroke to the feed plate, only six times the size. To reduce this to its

real size brought up another difficulty; the levers K W and W J were in a ratio of 6 to 7, so instead of reducing 6 to 1, it was necessary to combine the two, and make it  $\frac{36}{7}$ ; this is done in Fig. 4, A B being 6 inches and B C  $\frac{7}{36}$  of 6 inches, or  $1\frac{1}{4}$  inches. The distance O X spaced on Fig. 4 as A S, and the line S P, measured, gives the *actual* cam height for feed from hole 1 to 2. By inspection it is seen that this is the point or greatest feed of the curve, and deducting this height from the fixed outside diameter of the cam, gives the neutral circle N N, Fig. 2. This is the point on R of "no feed," and the line that we must work from. The cam circle is divided into thirty parts, and on these radial lines the distances, as S P, laid off, which have been determined on the "charts." Examining the cam, it will be seen that the curve crosses very easily the neutral circle, half way round, and the point D, Fig. 1, is on the cam where the very sharp drop is shown; this sends the block R on Fig. 2 from its lowest position beneath D to the position shown in the cut.

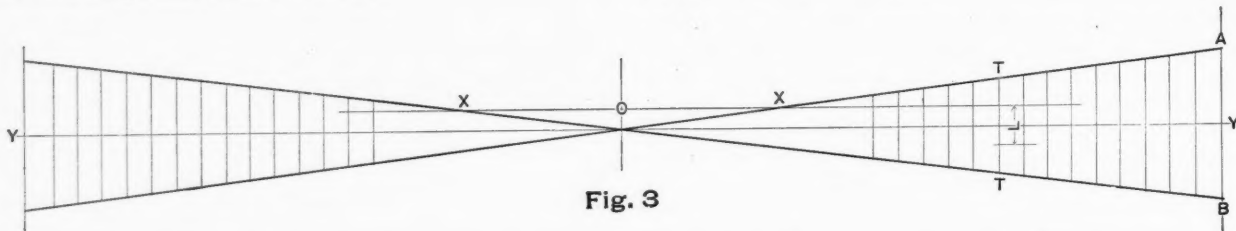


Fig. 3

The time was very short in laying out this cam, the dividers alone being used, and after eighteen had been accurately plotted (some of them with sharper reverses) without error, the system was pronounced a success. The accuracy might be questioned, theoretically, in several points; for instance, in Fig. 3, in drawing X X, the slightest variation would make a considerable difference in O X. This was balanced by measuring both sides of O and splitting the difference, the scale being 6 to 1. This change was not apparent on the work. It might be claimed that the vertical lines, Fig. 3, should be circular ones, but this was too small to notice. Then the distance L was divided on each side of Y Y, to diminish any injurious difference between the circle and its tangent.

An accurate drawing was made of the scales, after the paper was well stretched and glued to a board, and tracing paper used for subsequent operations. We soon found our original becoming well punctured, so when the great boom comes, an engraved plate will be furnished for this work.

This sketch shows only one of the many cases where this method can be applied, and which I have always found economical to adopt. Whenever a standard part is to be changed, a consultation with such a scale will give at once the proper new size and shape, which has been previously carefully worked out, giving sufficient strength, yet with economy in metal, and above all, maintaining uniformity of design and not showing the result of each draughtman's fancy.

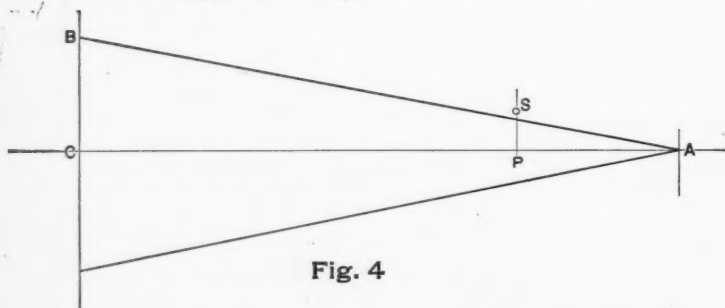


Fig. 4

Of course, the scale systems can be carried too far, creating confusion and errors, the very thing they are expected to prevent; but each establishment would have a different limit for their uses, but when employed with common sense and moderation, they are certainly very valuable.

\* \* \*

The catalog of the University of Wisconsin indicates a healthy growth of that institution. The list of professors and instructors is a long one and contains some well-known names.

\* \* \*

READERS of MACHINERY who wish to bind their copies may obtain an index for Volume III., free of expense, by sending a request to this office.

## THE SPLIT PULLEY PATENT AGAIN.

The short article of our last issue on the Split Pulley Patent has brought to us a letter from our valued contributor, Mr. R. D. O. Smith who is also the secretary and attorney of the Dodge Manufacturing Company, in which he very naturally takes the opposite view of the question. We desire to afford the Dodge Company the fullest opportunity to present their side to our readers, but it is impossible, from lack of space, to insert all of Mr. Smith's letter, which, we regret to say, is largely devoted to irrelevant subjects, such as the history and financial standing of the Chattanooga Company, the settlement with the Reeves Company, the assertion that parts of our article were false, if not libelous, etc.

Our position was and is that the essential claims of the patent were anticipated by the use of like devices for many years previous to its issue, and we have every reason to believe that the popular verdict of mechanics who are familiar with common shop

practice in this country is overwhelmingly to the same effect, notwithstanding the fact that the legal decision is, for the time being, against this conclusion.

The essential points of Mr. Smith's letter are given below:—

MISHAWAKA, Ind., July 6, 1897.

In your issue for July, '97, page 352, is an editorial entitled "The Split Pulley Patent," which contains matter which demands notice, because it is calculated, from the character of your journal, to do us direct and extensive injury and involve your readers in risks which, if properly informed, they would prefer to avoid. The substance of this objectionable matter has been going the rounds of the press, or certain lines of it. It emanated from the company whose defence you assume and it is not without surprise that journals like yours should fall into any trap of such setting, especially in view of the several corrections which have been published.

If your article contained nothing but the truth, it would not find any objection from us. We have not fought seven years to establish a lie, and it is not to be supposed that the U. S. courts would aid and abet us, if we had.

The litigation was commenced in 1889 and decided in 1896. The defendants were the Chattanooga Co., the Reeves Co. and the Keasey Co. They quickly pooled the issues and made common defense, so that their records are identical and comprise nearly 2,000 printed pages. Testimony referred to 26 anticipating uses, about 20 of which were identical with the exhibits which you parade as so conclusive against us. Yet in view of all this, Judge Sage said these alleged anticipations "do not impair, much less do they weaken the patent."

I enclose a sheet of cuts from the exhibits relied on by the defendants. You will find 22 of them are identical with the pulleys which you display with so much confidence and assure your readers that they conclusively show not only that "no one can lawfully claim any monopoly in the split pulley business at this late day," but also in particular that the Gilbert pulley in no way infringes the claims of the Dodge people, and 28 antedating split wooden pulleys.

I am confident if you had taken the proper means to gain information, you would have discovered in the first place that no one has made any claim to a monopoly of the split pulley business at this late day nor at any earlier day, and in the second place, that the old split pulleys—such as you show, and many of them quite as old—were the subject of testimony from many mechanical engineers whose character or ability you would not question.

Your paper assumed to occupy a high and impartial position. We have supposed its columns to be closed against all mercenary appeals, and we think such a position is incompatible with advice, which is to say, the least, calculated to throw discredit upon the decisions of the court and to encourage a disrespect for the law. That is not the way of good citizenship.

Respectfully, R. D. O. SMITH.

The two essential claims of the Dodge and Phillion patent are, as follows:—

1. A separable pulley, whereof, when the meeting ends of the rim are in contact, the meeting faces of the spoke-bar and hub are slightly separated, as described, combined with clamp bolts G, whereby said hub is clamped upon the shaft in the manner set forth.
3. A separable pulley whereof, when the meeting ends of the rim are in contact, the meeting faces of the spoke-bar are slightly separated,

and clamp bolts G combined with a separate split thimble interposed between said shaft and pulley, substantially as set forth.

It is perfectly evident that the pulleys illustrated and described in our last issue meet the requirements of the first claim exactly. About this there can be no dispute, and they clearly anticipate this count in the patent.

About the second count (the third claim of the patent) there may be a chance for an honest difference in opinion. Judge Sage, who rendered the decision, holds that the split bushings must be of wood, or its equivalent, for the reason that the specification of the patent states that the pulleys are to be of wood. He, however, curiously waives this point in two other particulars by stating:

First, that the construction referred to in the first claim is equally applicable to wooden and metallic pulleys, and, second, that the wood thimbles referred to in the third claim are equally applicable to metallic or wooden separable pulleys.

Our older mechanics can recall what they have seen used in the way of split bushings, of wood or other-

tightener or other device for adjusting the belt, and makes a smooth drive. The changes of cutter speed are obtained by shifting the belt from one cone to another. The cutters are set from a fixed point not subject to wear, and can always be set central with the work spindle, regardless of the wear of the cutter slide.

The feed has a wide range of evenly graded variations, and is driven by worm and worm wheel. The forward movement of

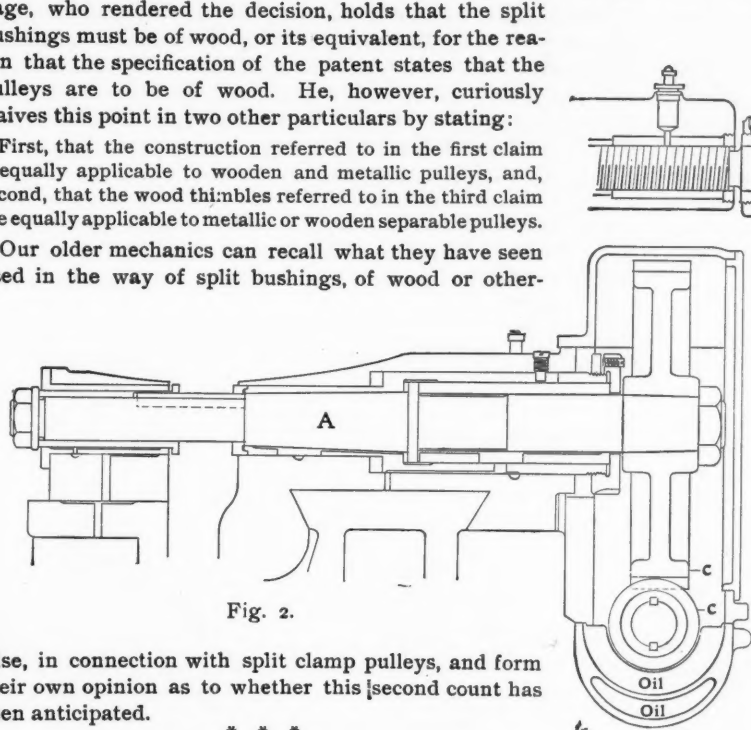


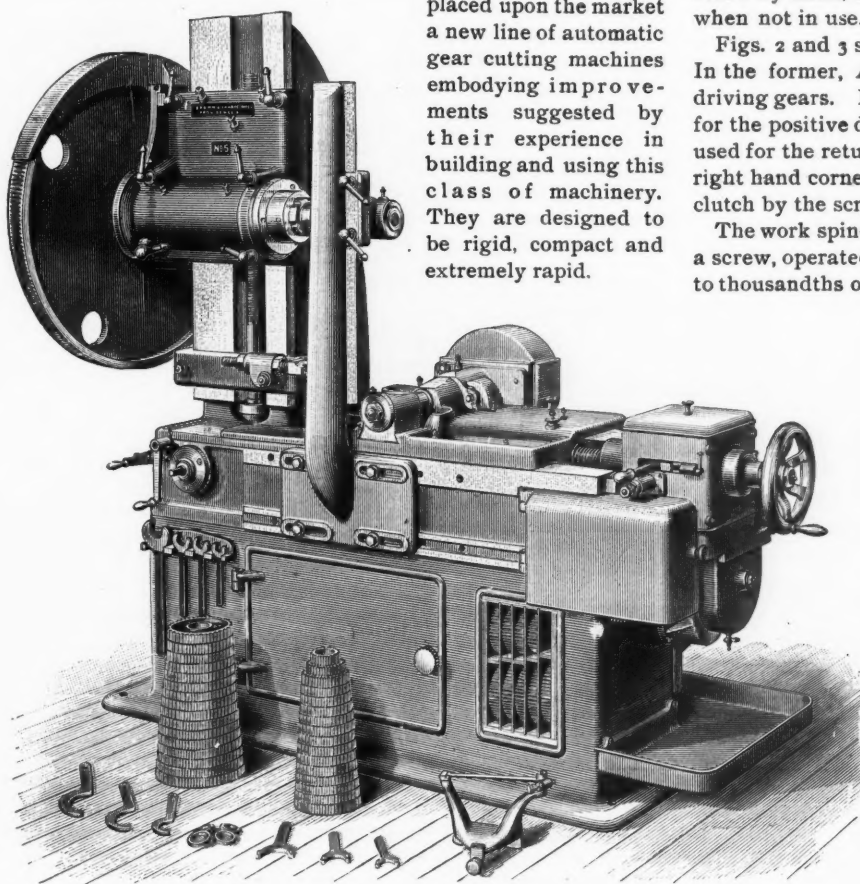
Fig. 2.

wise, in connection with split clamp pulleys, and form their own opinion as to whether this [second count has been anticipated.

\* \* \*

#### NEW AUTOMATIC GEAR CUTTING MACHINES.

The Brown & Sharpe Mfg. Co., of Providence, have recently placed upon the market a new line of automatic gear cutting machines embodying improvements suggested by their experience in building and using this class of machinery. They are designed to be rigid, compact and extremely rapid.



NEW AUTOMATIC GEAR CUTTING MACHINE.

The illustration shows one size of the machine and is typical of the different sizes.

The cutter spindle is supported on both sides of the cutter, and is driven by spiral gears, which does away with the belt-

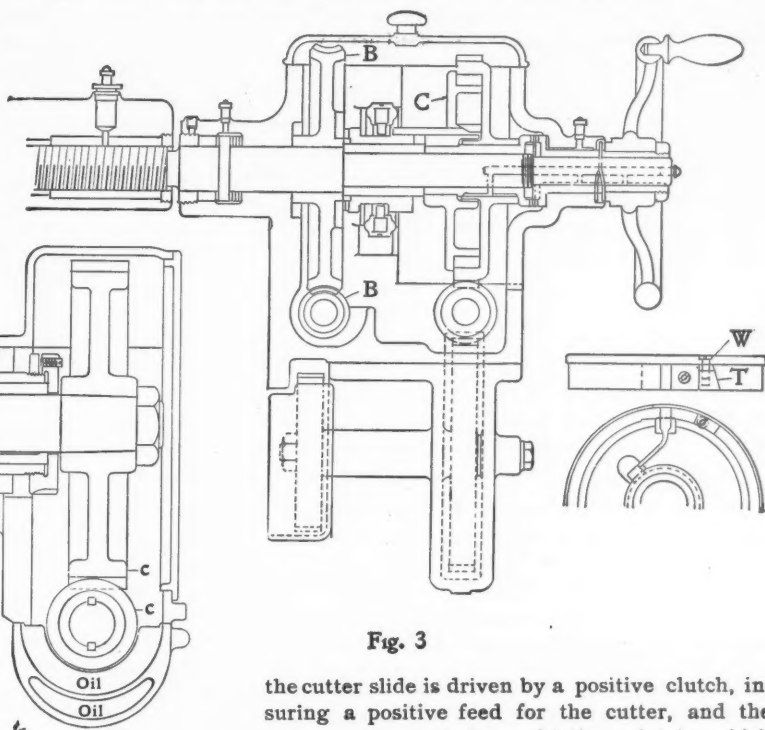


Fig. 3

the cutter slide is driven by a positive clutch, insuring a positive feed for the cutter, and the return movement by a friction clutch, which receives the shock of the reversal and thus reduces the strain on the feed mechanism. The hand wheel, for operating the feed screw by hand, disconnects automatically and remains stationary when not in use.

Figs. 2 and 3 show details of the drive and feed respectively. In the former, A is the cutter spindle and C C are the spiral driving gears. In the latter, B B are the worm and worm wheel for the positive drive during the cut, and C is the friction clutch used for the return. This is shown more in detail in the lower right hand corner of Fig. 3, where the method for adjusting the clutch by the screw W, and wedge T, is indicated.

The work spindle slide is moved up and down on the upright by a screw, operated from the front of the machine, and a dial reading to thousandths of an inch enables the depth of cut to be accurately determined. The larger machines are furnished with outer supports for the work spindle.

Great care is taken to have the index wheel very accurate, and it is large in diameter in proportion to the diameter of the work. The indexing mechanism is unusually rapid and error in its action is practically impossible. When the machine is cutting, the indexing mechanism is at rest and not subject to the strains of a constantly revolving friction. The drive for returning the cutter slide and for indexing is independent of cutter drive and feed, and the return of cutter slide and speed of indexing mechanism are the same, whatever the speed or feed of cutter.

Provision is made to allow for re-cutting or setting a gear without loosening the gear on the arbor or the arbor on the spindle, or slipping the teeth of the change gears.

A withdrawing expansion arbor is furnished with the larger machines, and can be drawn back through the work spindle, allowing a finished gear to be removed and the blank spaced without disturbing any of the adjustments of

the outer support or of the work spindle slide.

The driving gears and worms run in oil, and receptacles are provided inside the base of the machine for catching the chips and for holding lubricant. The chips fall directly from the cutter

into the receptacle, without the use of any automatic device for the purpose, and when using a pump, the lubricant is taken from and flows back to the reservoir in the base, strainers being provided.

\* \* \*

JUST what is best for journals and bearings is one of the things that seriously puzzle the man who starts out to design machines and machinery. Morin, who did a good deal in the way of friction and the like, never settled it to his own liking, and of course not to the liking of anyone else. There was, perhaps, never a question over the superiority of the old-time hammered steel shaft. There is some question as between the steel shaft of commerce of the present day and a good hammered iron shaft. That question was raised fifteen years ago, and has not been settled up to the present writing. "All steel shafts" used to have a good deal to do with the matter when the steel cost the most money. There was something to talk about. It has lost this advantage.

\* \* \*

### HOW AND WHY.

A COLUMN INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST. GIVE ALL DETAILS AND YOUR NAME AND ADDRESS, WHICH WILL NOT BE PUBLISHED UNLESS DESIRED.

42. J. S. H. Please tell me a simple way of finding the dimensions of a worm to be cut in a lathe. The wheels are generally cast instead of being cut, and as the drawings give us only the pitch, pitch circle and diameter of the worm, we have no way of finding out how to shape a tool to cut the thread at the right angle. *A.* The drawings ought to show a section of the worm parallel with its axis and give the dimensions of the teeth shown in section. Without this you have no means of knowing how to shape the tool. Worm gearing is generally constructed so that a tool with straight sides and an angle of about  $30^\circ$  between them, will be right to use. This may or may not be correct in your case.

43. H. A. C. writes: Please give me a reliable formula for a solution for coating blue-print paper. *A.* Blue-print paper can now be bought coated, ready for use, from any dealer in drawing materials, and most draughtsmen prefer to use it, as it is more convenient. In case you do not have much printing to do, however, you may prefer to prepare your own paper. The following solution gives good results:

Red prussiate potash (recrystallized)... 1 part (by weight).  
Citrate of iron and ammonia..... 1 " "  
Water..... 10 " "

In mixing the solution, first weigh and pulverize the potash and dissolve it in the required weight of water, afterwards add the iron, which does not need pulverizing. If the chemicals are dissolved in separate bottles of water they will keep better than if mixed, as the light will not affect them. 2. What is the best paper to use for blue-printing? The solution soaks into the paper that I have and makes spots. *A.* "Helios" paper, for sale by the Keuffel & Esser Co., New York, gives good results. It comes in rolls of various widths and thicknesses. To prevent spotting, the paper must be coated quickly and hung up to dry.

44. F.M.A. writes: I have seen so much printed on the safety-valve and the proportions of the same that I want to study into the question for myself. Kindly inform me how high an ordinary weighted valve will lift. *A.* From .05 to .10 of an inch; the higher the pressure the lower the lift. 2. Is there a good rule for calculating the weight of steam that will flow out? *A.* To get flow in pounds per second, add 15 to the gauge pressure, in pounds, multiply by the area in inches and divide by 70. Note that the area meant is not the area of the valve, but the area of the opening to the air, which is equal to the circumference of the valve multiplied by the lift. For example, let the boiler pressure be 60 pounds and area of opening two square inches. Then  $60 + 15 = 75$ ;  $75 \times 2 = 150$ ;  $150 \div 70 = 2.14$  pounds per second. This is Napier's rule for the flow of steam, and one of the best.

45. C. F. asks: How is the economy of gas engines affected by a change in load? Does an under-loaded engine show a loss in economy the same as in the case of a steam engine? *A.* Yes. A report of a test of a 12 HP. gas engine has been furnished us which gives the following results: One HP., 48 cu. ft. of gas per HP. per hour; 6 HP., 18 cu. ft.; 8 HP., 16 cu. ft.; 12 HP., 15 cu. ft.

### WHAT MECHANICS THINK.

THIS COLUMN IS OPEN FOR THE EXPRESSION OF PRACTICAL IDEAS OF INTEREST, TECHNICAL OR OTHERWISE. WRITE ON ONE SIDE OF THE PAPER ONLY, AND BOIL IT DOWN.

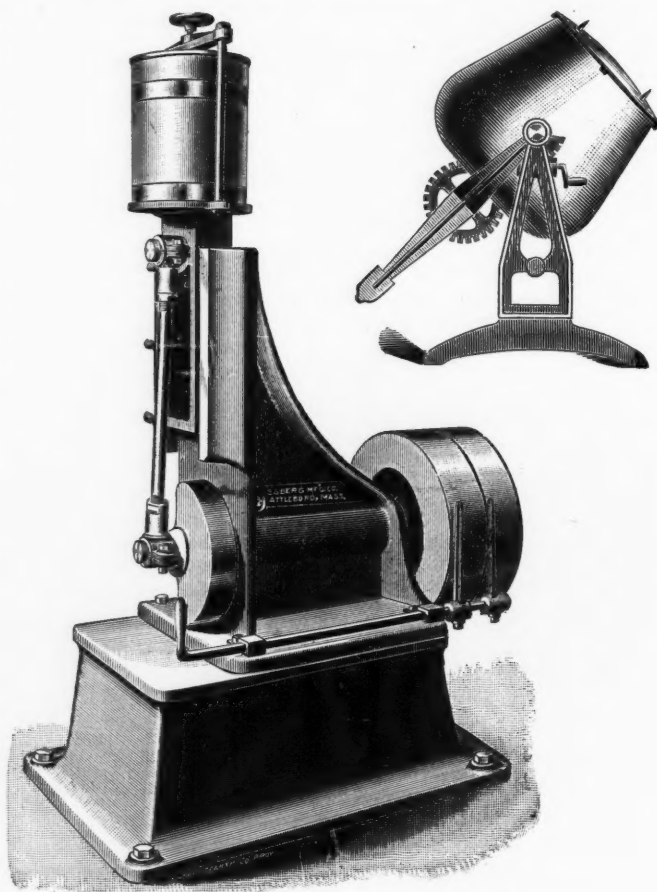
WHEN SKETCHES ARE NECESSARY TO ILLUSTRATE THE IDEA, SEND THEM ALONG—NO MATTER HOW ROUGH THEY MAY BE, WE WILL SEE THAT THEY ARE PROPERLY REPRODUCED.

For the best practical idea sent us each month to use in this column we will send a copy of Usher's "Modern Machinist," or Grimshaw's "Shop Kinks." For the next best we will enter a subscription for one year to MACHINERY to any desired address. For the next best, or for any idea that is worth publishing, we will send a copy of Colvin and Cheney's "Machine Shop Arithmetic."

### FINISHING PRESS WORK.

Perhaps a few hints on the best method of finishing up punched work may be of interest to some of your readers who have the good or bad fortune to have charge of that class of work.

Some years ago I took charge of a large press room, doing a great variety of both iron and brass work, and one of the first things the superintendent said to me was: "I don't know how it is, but we have never been able to do press work equal to that which we buy from outside. We hire the best tool-makers we can get, and take a good deal of care in fitting our punches and dies, but somehow the work never seems to come out so good as it ought to, and I wish you to see if you can find out why."



Well, I started in, and before night found out what the matter was, and sent a requisition to the office for a tumbling barrel like the one shown in the first sketch. I got called to the office on it at once. Said they didn't see what I wanted of it, but finally they were prevailed upon to order it. The trouble was the work was being sent out of the press room just as it came from the dies, all grease and dirt and more or less scrap mixed with the same. The tumbling barrel arrived and was set up, and pretty soon the press room began to get lots of compliments which was not a common occurrence in the factory. It wasn't that kind of a shop.

The best thing to use for finishing work with the barrel is old broken emery wheels, but when the supply of these runs short, a few cinders from the ash pile will do nearly as well. Of course, a liberal supply of good hard-wood saw-dust is also essential.

The cut does not show up the barrel as well as it should. The

machine is driven by a pulley with a gear on the hub which meshes into the gear seen on the further side of the barrel. The small crank shown is for giving the tub a different incline or for emptying it when the work is finished.

The machine shown in the second cut, and which, by the way, will be new to many of your readers, is called a tubbing machine, and is used for practically the same purpose, except that it is better adapted for work that is not easily bent, such as bicycle links, washers, or any kind of work that will stand rough usage. It is no good for long, delicate punchings. For that class of work the best machine is one like the inclined tumbler shown in the first cut, as you can give it any incline you wish, and the work will just roll round and not fall and bruise itself. For work that you would like to put a nice finish onto, such as pins, clock movements, or the cheaper grades of jewelry, a thick heavy soap suds will be found very serviceable, first tumbling the work well in the soap suds, and then in hot sawdust to dry the pieces.

A power sieve for separating the sawdust and the work each being run off in opposite directions by means of two chutes properly arranged, is also a great help in turning out good press work, and I would be pleased to send a sketch to any one interested.

J. L. LUCAS.

## SILVERING COPPER.

The following recipe for silvering copper plate was found in an old note book, and might be of interest to some of your readers.

Apply a solution of nitric acid and water of equal parts, with a rag on a stick, then apply some quicksilver and wipe it all over the plate until it is completely silvered. It is said that small particles of gold dropped upon this surface will adhere to it as iron filings to a magnet, and is used by miners and others to separate gold-dust from other material.

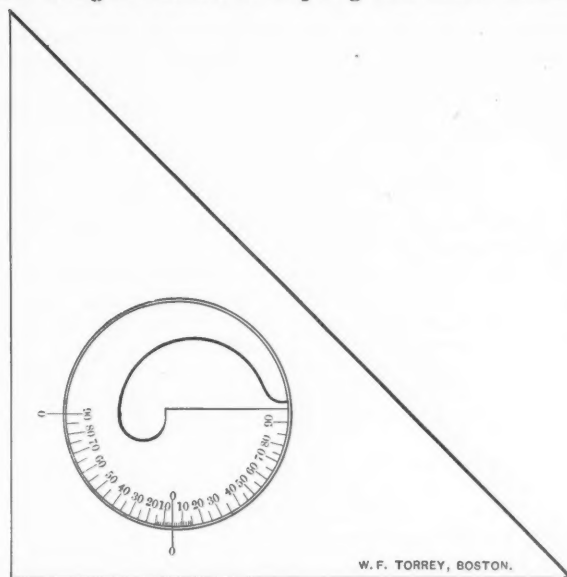
Like most mechanics, I haven't got gold enough to try the scheme, so I will submit it to those who may be more fortunate.

"A. HARDCASE."

## ANOTHER PROTRACTOR.

Noticing the Protractor in your May issue, I thought perhaps the one shown herewith would interest your readers, the new

feature being the absence of a *spring* and that it has an *arm*



which will allow of its being sprung in place. There being only two parts, the cost of getting them up is cheapened.

Medford Hillside, Mass.

WM. F. TORREY.

[We understand that an application for a patent has been made.—ED.]

\* \* \*

## MANUFACTURERS' NOTES.

A. D. QUINT, Hartford, Conn., manufacturer of turret drills, reports that during the month of July exports of his turret drills were made to Germany and Russia. The shipments to parties in this country made a total of 28 spindles, fitted with the reversing tap holders, including one of five drills to the Davis & Farber Machine Co., North Andover, Mass.

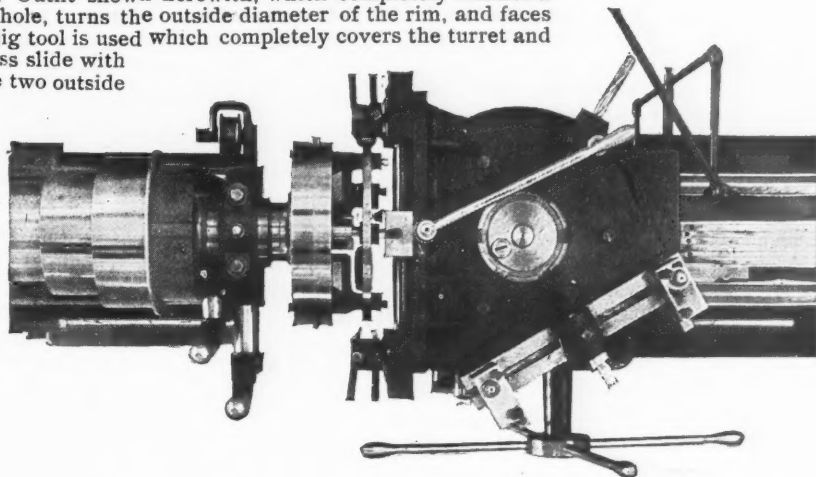
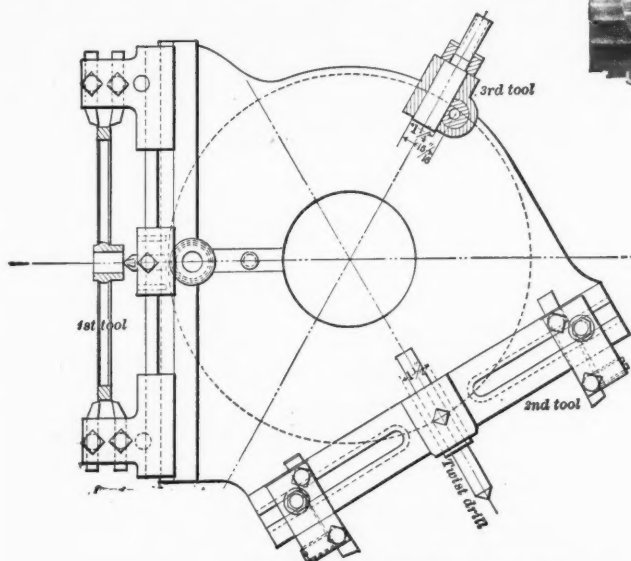
THE RUE MANUFACTURING Co, whose office has been so long located at 116 North 9th street, Philadelphia, Pa., have removed their office to

See preceding issues of MACHINERY for detailed descriptions.

## A New Lathe System—VII.

One of the adjuncts of our system is the Gear Outfit shown herewith, which completely finishes a gear blank at the first setting, bores and reams the hole, turns the outside diameter of the rim, and faces both edges of the rim and hub. For this purpose a jig tool is used which completely covers the turret and is firmly bolted and doweled to it. It carries a cross slide with three tool-blocks, which move simultaneously. The two outside blocks each carry two tools; the middle block carries only one. The outside blocks rough and finish the sides of the rim, while the middle block faces the hub.

The second position carries a drill and a roughing and finishing tool for turning the outside diameter of the rim. At the third



position a reamer and counterbore are located. The inside face of the hub is finished by a facing cutter held in the end of the dead spindle; the horizontal lever shown near the roller feed operates this spindle.

When the gears are finished, the fixture and tools can be placed on a shelf in the tool room, without disturbing the setting, and another fixture with tools set for another piece of work bolted to the turret; or the machine can be used as a screw machine on regular work. By means of this jig system, the machine can be set up, ready for work, in a few minutes, and is thus very efficient when only a few parts are to be made at a time. We make similar outfits for other small cast iron work. These illustrations of course represent only one adjunct of our labor-and-money-saving system. Possibly it does not apply to your business; but if you are producing lathe work in even moderate sized lots we can probably save you money. An enquiry will cost you nothing.

**JONES & LAMSON MACHINE CO., SPRINGFIELD, VERMONT, U. S. A.**

FOREIGN REPRESENTATIVES—HENRY KELLEY & Co., 26 Pall Mall, Manchester, England.

M. KOEYEMANN, Charlottenstrasse, 112 Dusseldorf, Germany; representative for Germany, Belgium, Holland, Switzerland and Austria-Hungary.

their factory building, at 215 Race street, (same city), in order to facilitate the transaction of business and avoid delay in repair shop.

MR. CHAS. DAVIS, president of the Davis & Egan Machine Tool Co., has just returned from an extended trip, and in recognition of the foreign business which he has brought to Cincinnati, the leading commercial bodies of that city tendered him a banquet at the Queen City Club a few nights ago.

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### FRESH FROM THE PRESS.

*The Design and Construction of Modern Steam Engines.*  
Theo. F. Scheffler, Jr., Erie, Pa.

Mr. Scheffler, who has been one of our frequent contributors, is preparing a series of blue-prints containing drawings and descriptive matter relating to the design and construction of modern, non-condensing engines. These prints are issued in about 30 parts of 12 plates each, size 9x12 inches. We have received copies of the first three sections, which are well executed and contain full and clear explanations of the methods of procedure in designing, with references also to shop methods.

The subject is treated by taking up one size of engine and following through with all the calculations relating to it. After a person understands these calculations, he should be able to carry them through for any other size with the aid of the notes and tables that are furnished. This method we believe to be a good one.

The author informs us as follows: "It is my intention to make this book a complete guide for designing non-condensing engines. The book deals only with the simple forms of engines, with the exception of the valve gear, which has been designed for independent steam and exhaust valves, for the reason that so much has been written about engines with simple 'D' valves, and nothing, comparatively speaking, about four-valve engines."

We judge that the work does not discuss engines larger than 400 or 500 HP., and that it applies mainly to engines with shaft governors. The price of the separate sections is 30 cents each, sent to any address in the United States or Canada.

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### ADVERTISING LITERATURE.

THE STANDARD SIZES FOR CATALOGS ARE 9x12, 6x9 AND 3½x6 INCHES. THE 6x9 IS RECOMMENDED, AS THIS SIZE IS MOST LIKELY TO BE PRESERVED.

NEW BRITAIN MACHINE CO., New Britain, Conn. Catalog of chain saw mortiser.

This contains some very good illustrations and interesting description of the chain saw mortiser made by this company, which we have mentioned before. The cutting edges are really links of an endless chain which travels at the rate of 1500 feet per minute, presenting 40000 teeth to the work in this space of time. Woodworkers will find this interesting.

P. F. OLDS & SON, Lansing, Mich. Catalog of gas and gasoline engines.

Those interested in this form of engine, and they are daily increasing, will find much to read and ponder in this catalog. Simplicity seems to be a leading feature, and is a point which is rarely overdone. They are apparently made for every service, from a threshing machine to a yacht, and the testimonials speak highly of them.

THE WATERBURY FARREL FOUNDRY & MACHINE CO., Waterbury, Conn., have issued the next of their series of catalogues, E, devoted to foot presses, drop presses and screw presses, which contain in compact form a great deal of useful information on these appliances. The catalogue, like all others issued by this company, is an excellent specimen of paper, engraving and typography.

PENNSYLVANIA BOLT AND NUT CO., Lebanon.

Besides the usual line of bolts, nuts, rivets, etc., this company makes taps, forgings of various descriptions, and rolls of bar iron. The catalog is well bound in cloth and is a convenient reference-book in the above lines. It contains a few useful tables relating to weights and dimensions of bolts and screw threads. Size, 6x9 inches.

BRADY MANUFACTURING CO., Derby, Conn.

This well known company is now located at Derby, Conn., and have issued a little folder setting forth their facilities for manufacturing or experimenting with new devices. They have had a long experience in this work and have developed some of the best known specialties on the market.

ROLLINS ENGINE CO., Nashua, N. H. Catalog of the Rollins engine.

This is a thoroughly up-to-date catalog, Jensen old style type and all. The illustrations are good, show the important details of the engine in good shape, and is very interesting to those dealing with the steam engine. Their new valve gear is described and directions are given for adjusting it, which will of course interest the engineer who wishes to be abreast of the times. Several indicator cards from various engines in service aid the testimonials in saying a good word for the Rollins.

CLAYTON AIR COMPRESSOR WORKS, Brooklyn and New York. Catalog No. 9 of air compressors and compressed air appliances.

In addition to a list and description of the different patterns of compressors made by this company, the catalog describes various appliances for the use of compressed air, including apparatus for deep well and other pumping. It contains an illustrated descriptive article upon the "Widening Use of Compressed Air," part of which was originally published in the *Engineering Magazine*, and two tables, one of losses by friction and the other of losses by altitude in using compressed air. Size, 6¼ x 9½ inches; 106 pages.

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### POINTERS ON PUMPS.—VIII.

The Marsh deep well pumping engine, illustrated herewith, is intended to take the place of windmills and other cumbersome and expensive contrivances for pumping water from deep, or shallow wells, for farms, factories, dwellings, etc.

It is self-regulating in every way, and is the cheapest and most economical device for the purpose yet offered. It can be attached to any plunger rod; will elevate water from any depth and do it constantly without any attention whatever.

These engines have been extensively introduced in the natural gas regions, and work equally as well with direct pressure of gas as with steam.

The Marsh deep well engines are now made in five sizes, viz.: 4x6, 4x8, 6x12, 7x24 and 8x36, and in connection with suitable water cylinders will meet almost any ordinary requirement. Information or suggestions will be cheerfully furnished. In writing us in regard to same please state:

1. Size well casing (inside) and depth.
2. Distance to water from surface of ground.
3. Height and distance water must be elevated above surface of ground.
4. Capacity required in gallons per hour.
5. Steam pressure on boilers and distance of same from well.

THE BATTLE CREEK STEAM PUMP CO., Battle Creek, Mich.—Adv.

LIGHT SHEET METAL STAMPING, and small articles in that line produced at bottom prices. Write for estimate.  
tf FRANK WHEELER & SON, Meriden, Conn.

"MODERN STEAM ENGINE CONSTRUCTION."  
Parts 1 to 10, ready. Thirty cents each. tf  
THEO. F. SCHEFFLER, JR., 943 East 21st Street, Erie, Pa.

FOR SALE.—Blue Prints of the working drawings of plain slide valve stationary engines, in sizes from 1 to 40 HP. The latest and best, at a reasonable price. Let me send table of screw threads and decimal equivalents—it's handy.  
5t 4-1 OTTO E. EVANS, York, Neb.

WANTED—A practical mechanic as working foreman in a small shop. Address, tf  
"Hartford," care of MACHINERY, 411-13 Pearl St., New York.

